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AND

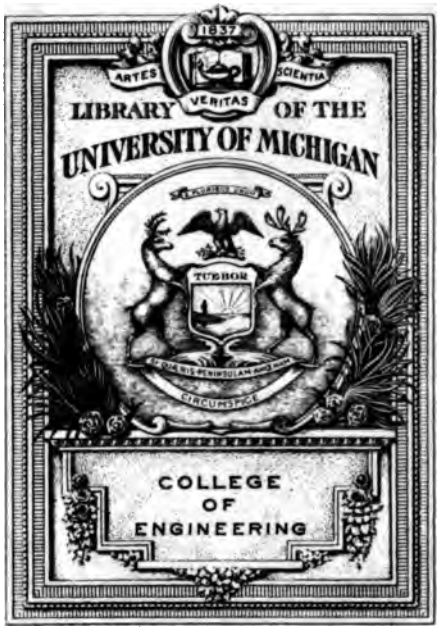
AUXILIARY MACHINERY

1914

BARTON AND STICKNEY

SCREW PROPELLER DESIGNING

DYSON



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PLATES

NAVAL RECIPROCATING ENGINES

AND

AUXILIARY MACHINERY

A TEXT-BOOK FOR THE INSTRUCTION OF MIDSHIPMEN

AT THE

U. S. NAVAL ACADEMY

BY

JOHN K. BARTON

Commander, U. S. Navy

Former Head of the Department of Marine Engineering and Naval Construction

REVISED

BY

H. O. STICKNEY

Commander, U. S. Navy

Recently Head of the Department of Marine Engineering and Naval Construction

AND

SCREW PROPELLER DESIGNING

BY

C. W. DYSON

Captain, U. S. Navy

ANNAPOLIS, MD.

THE UNITED STATES NAVAL INSTITUTE

1914

PLATES

NAVAL RECIPROCATING ENGINES
AND
AUXILIARY MACHINERY

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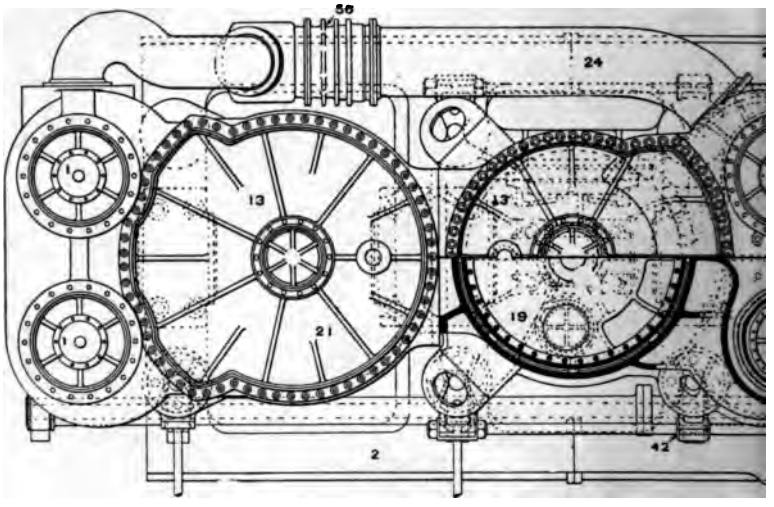
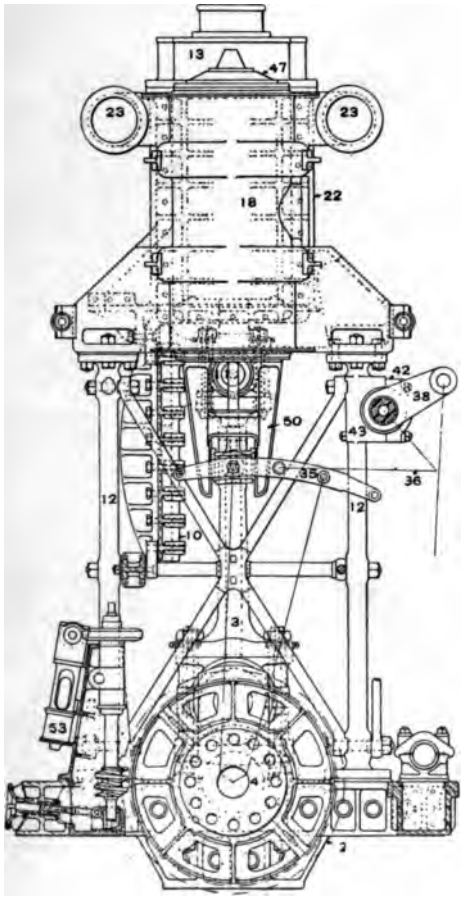
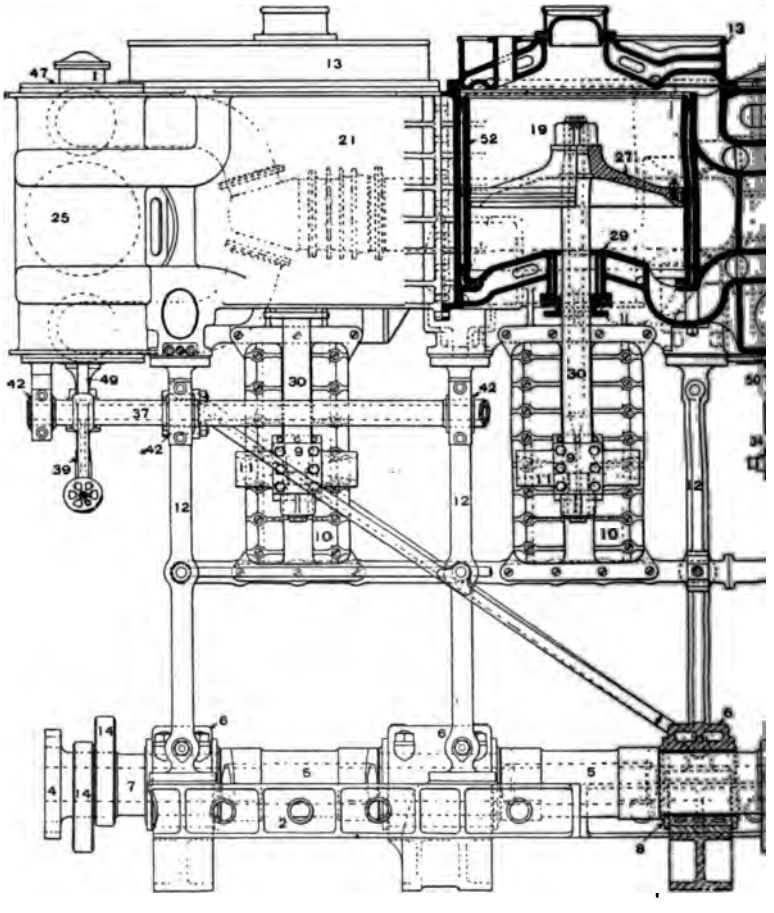
KEY.

1. Balance pistons.
2. Bed plate.
3. Connecting rod.
4. Couplings.
5. Crank pins.
6. Crank shaft bearings.
7. Crank shaft.
8. Crank shaft brasses.
9. Crossheads.
10. Crosshead guides.
11. Crosshead pins.
12. Columns.
13. Cylinder covers.
14. Eccentrics.
15. Eccentric rod, go-ahead.
16. Eccentric rod, astern.
17. Eccentric straps.
18. High-pressure cylinder.
19. Intermediate-pressure cylinder.
20. Forward low-pressure cylinder.
21. After low-pressure cylinder.
22. Main steam.
23. High-pressure exhaust or first receiver.
24. Intermediate-pressure exhaust or second receiver.
25. Low-pressure exhaust.
26. Piston, high-pressure.
27. Piston, intermediate-pressure.
28. Piston, forward low-pressure.

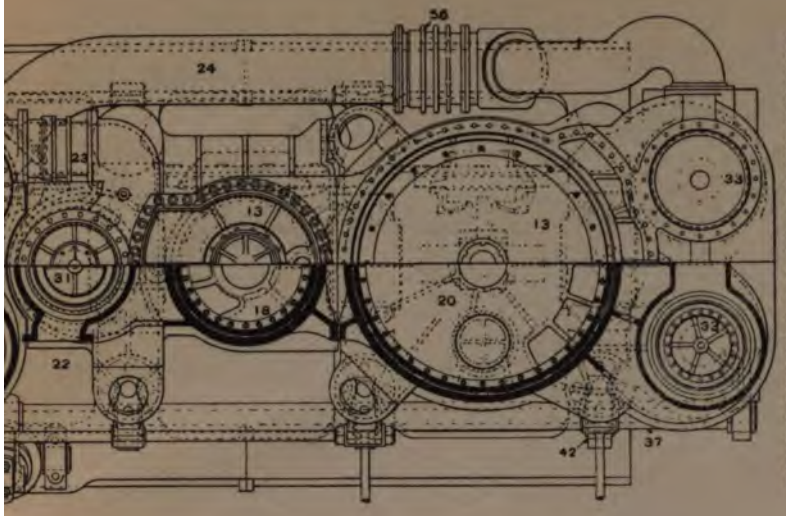




H. P. Cylinder, Looking Forward.

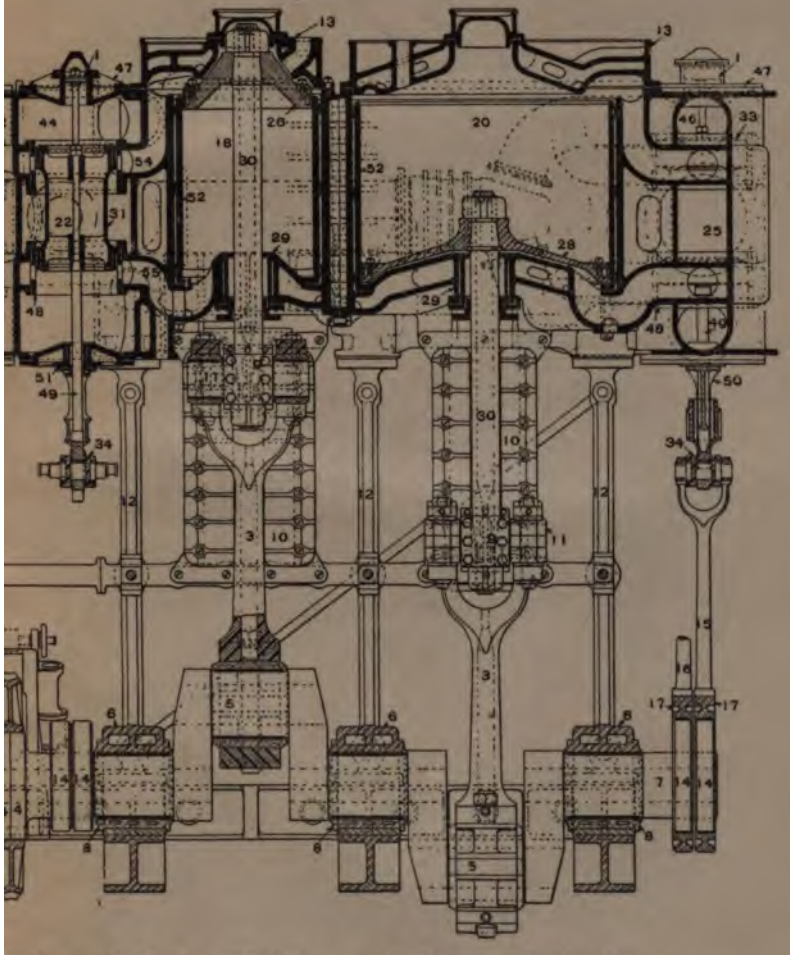


Four-Cylinder Inverted, Direct-Acting Triple-Expansion Engi

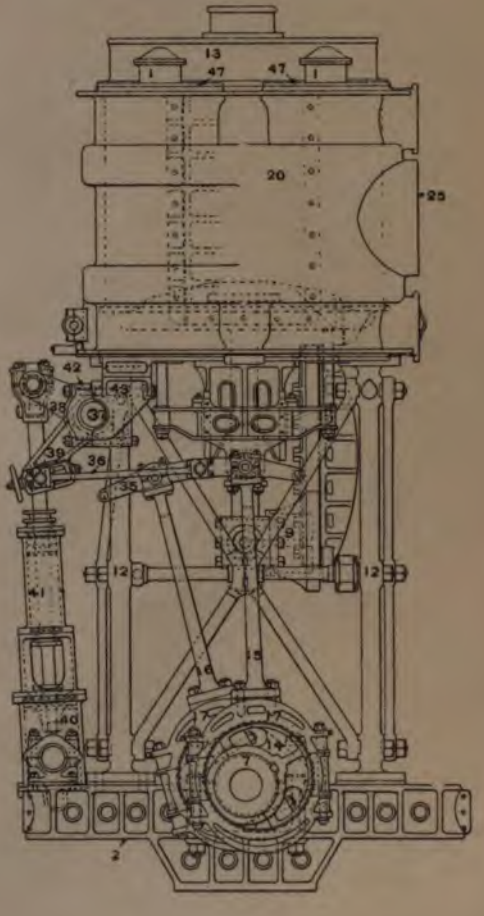


KEY.—Continued.

29. Piston rod stuffing boxes.
30. Piston rods.
31. Piston valve, high-pressure.
32. Piston valves, intermediate-pressure.
33. Piston valves, forward low-pressure.
34. Link blocks.
35. Main links.
36. Suspension links.
37. Reversing shaft.
38. Reversing shaft arm for reversing engine.
39. Reversing shaft arm for links.
40. Steam reversing cylinder.
41. Oil controlling cylinder.
42. Reversing shaft bearings.
43. Reversing shaft brackets.
44. Valve chest, high-pressure cylinder.
45. Valve chest, intermediate-pressure cylinder.
46. Valve chest, forward low-pressure cylinder.
47. Valve chest covers.
48. Valve chest liners.
49. Valve stems.
50. Valve stem guides.
51. Valve stem stuffing boxes.
52. Cylinder liners.
53. Turning engine.
54. High-pressure steam port, top.
55. High-pressure steam port, bottom.
56. Expansion joints.

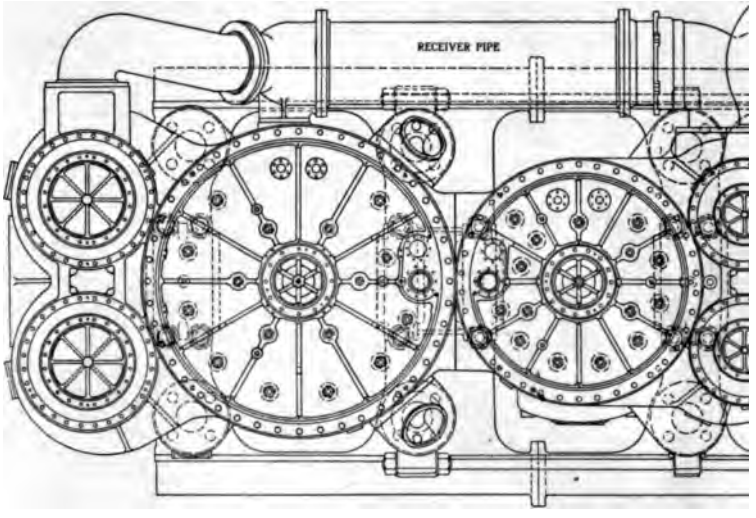


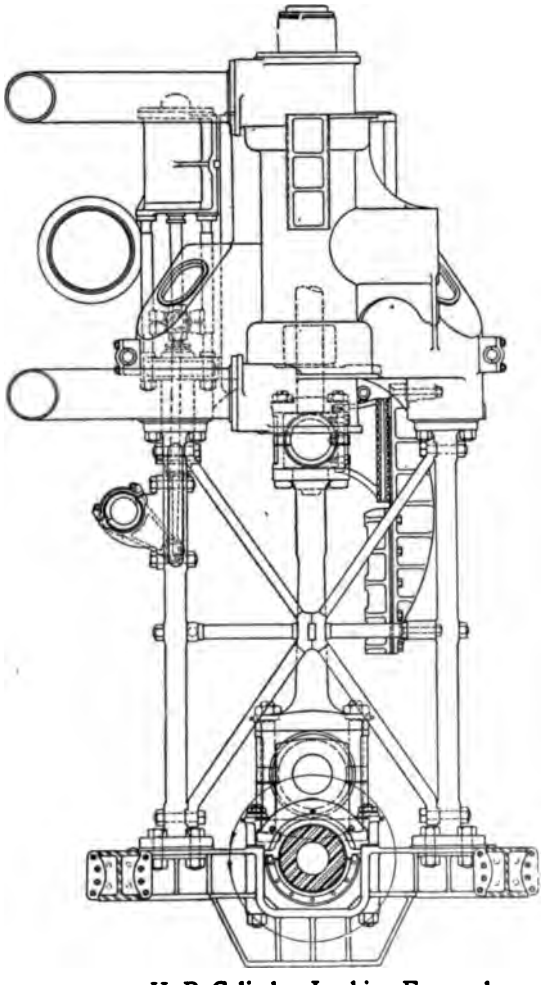
S. Armored Cruisers Washington and Tennessee Designed 1902.



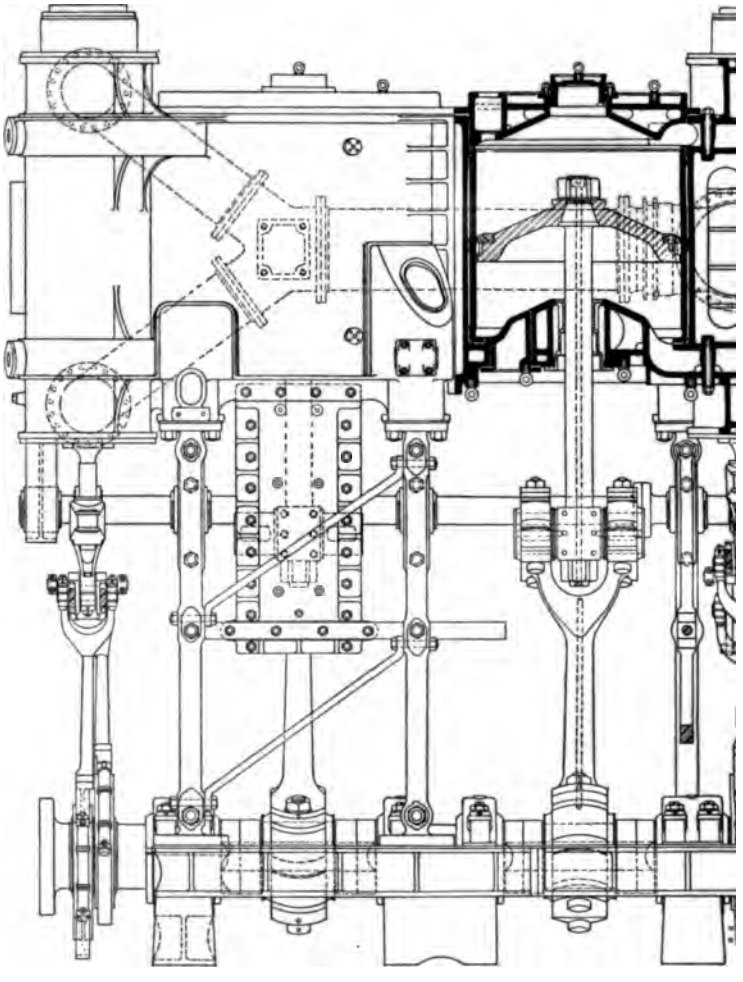
Forward L. P. Cylinder, Looking Aft.

Compare this design with that of the Tennessee class, Plates 1 and 2, designed in 1902. Note the longer piston valves, and the shorter and straighter port passages which decrease the volumetric clearance

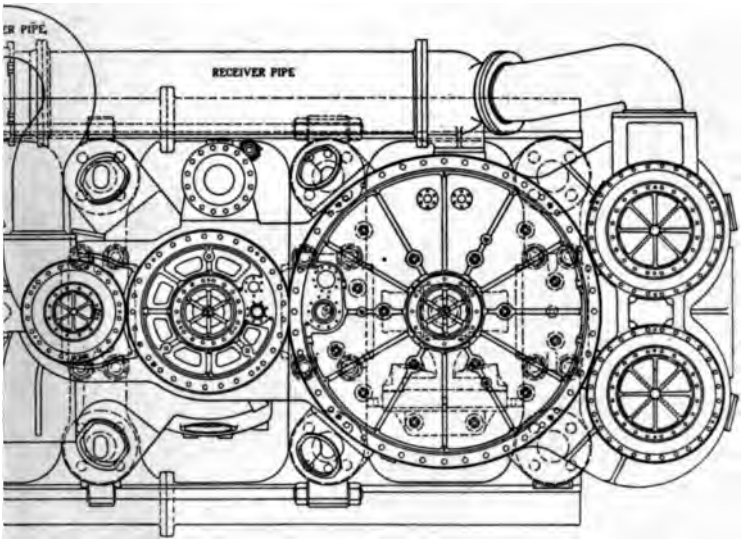


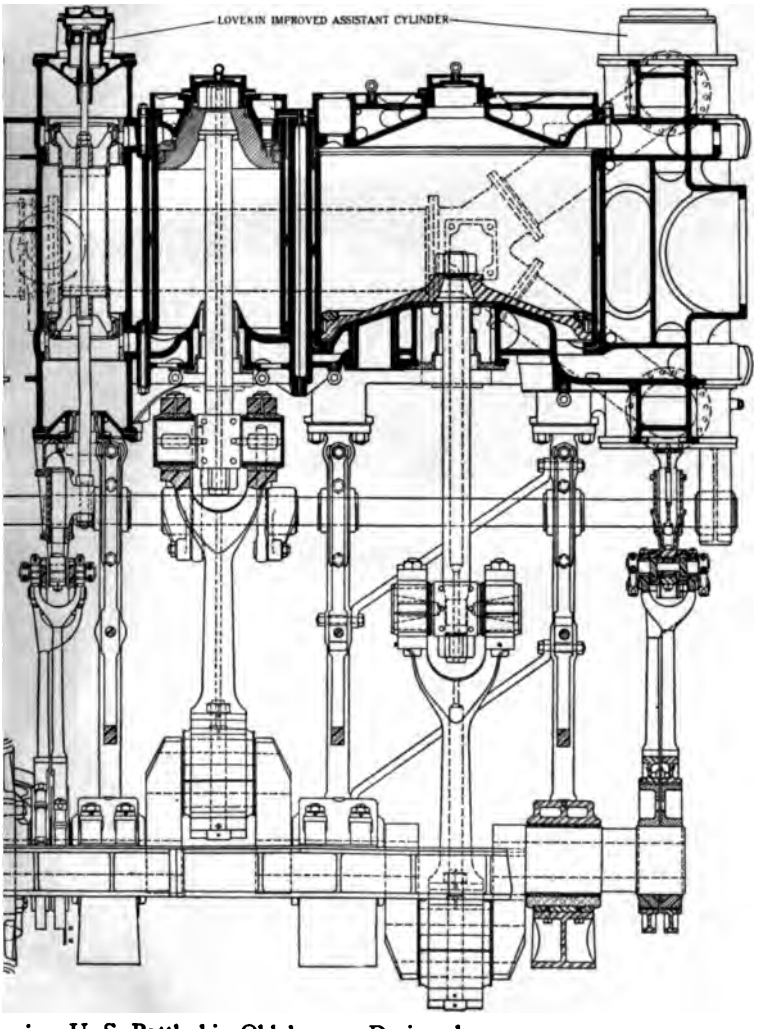


H. P. Cylinder, Looking Forward.

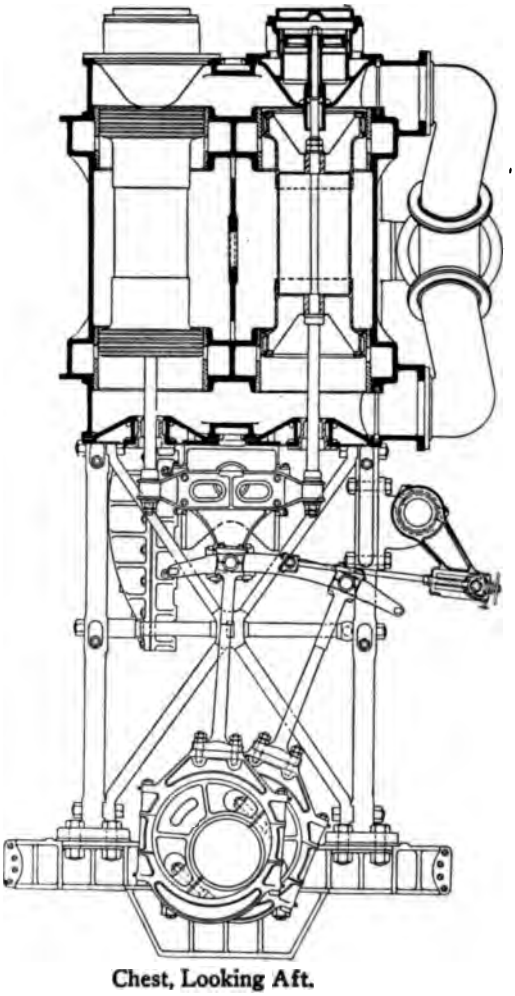


Four-Cylinder Inverted, Direct-Acting Triple-Expan





ngine, U. S. Battleship Oklahoma. Designed 1911.



Chest, Looking Aft.
Section Through Forward L. P. Valve

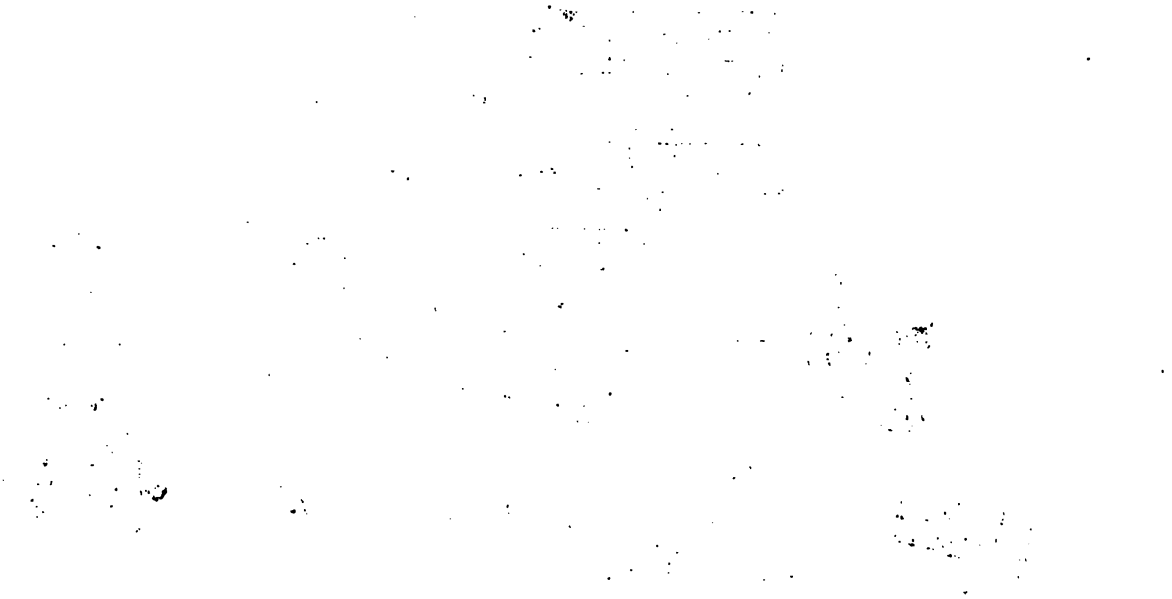
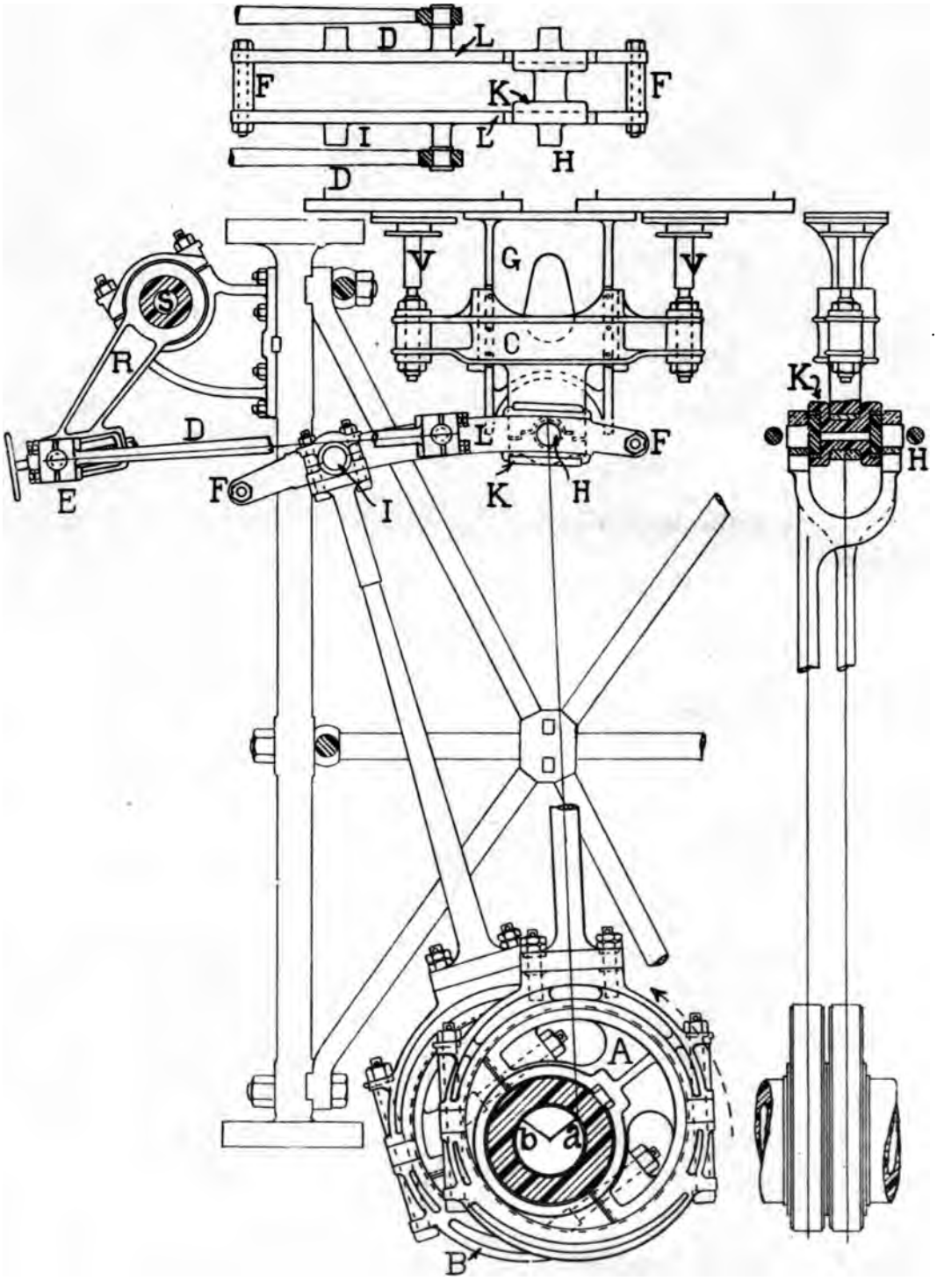
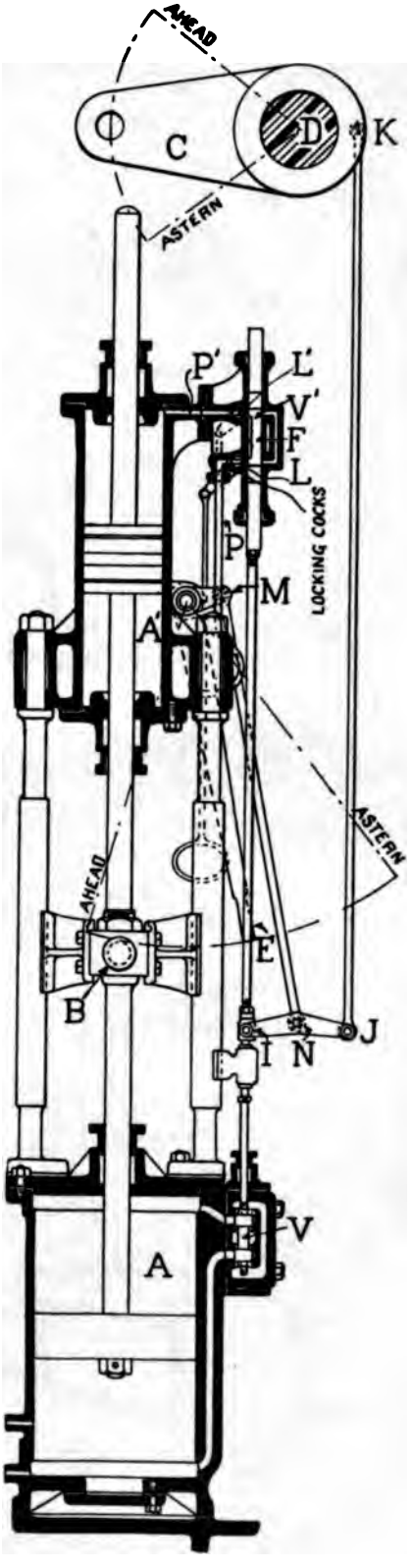


PLATE 6.




AHEAD
C
D
K
ASTERN
P'
L'
V'
F
L
LOCKING COCKS
P
M
A'
ASTERN
AHEAD
E
B
J
I
N
A
V

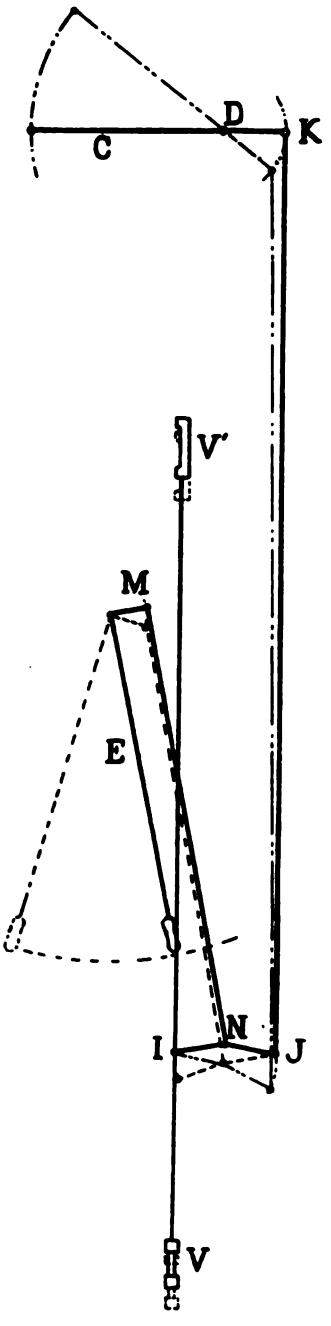

D
K
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PLATE 7.

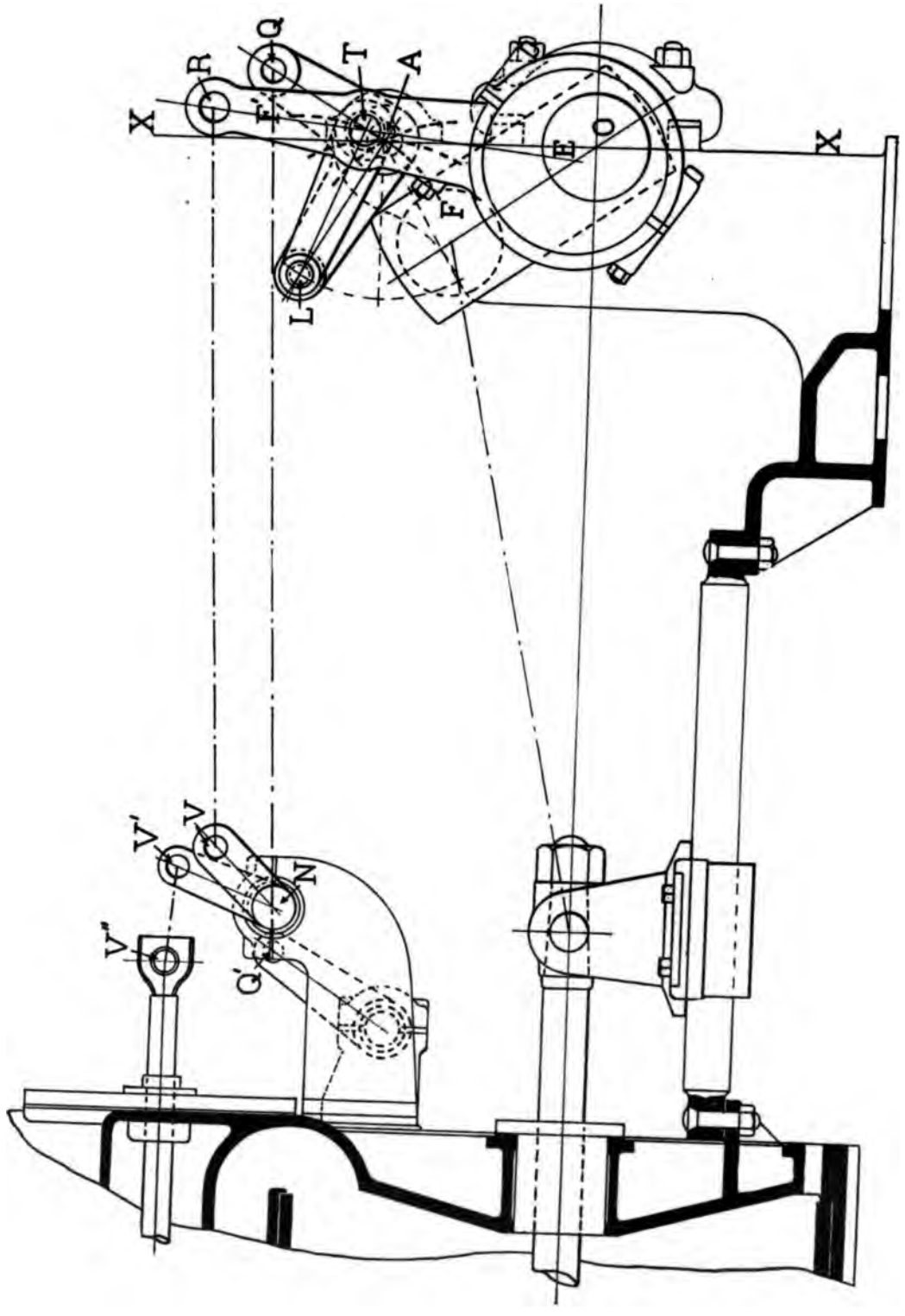

Fig. 1.

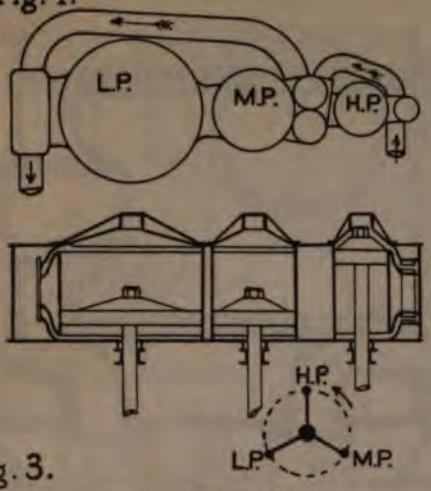

Fig. 2.

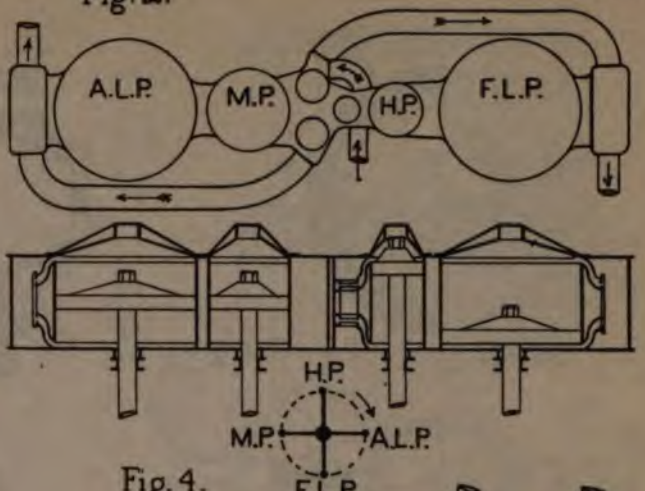

Fig. 3.

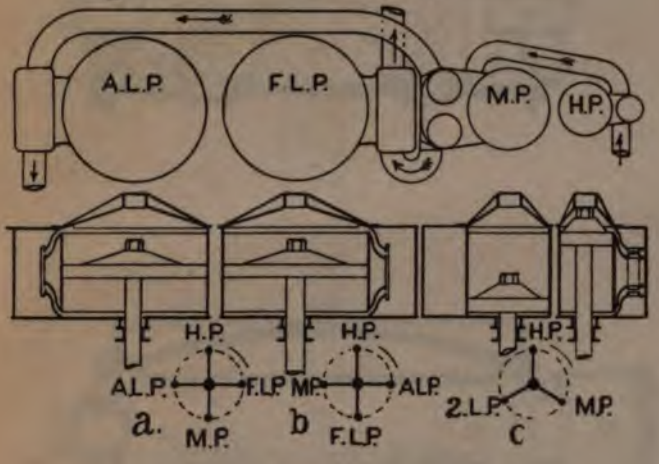

Fig. 4.

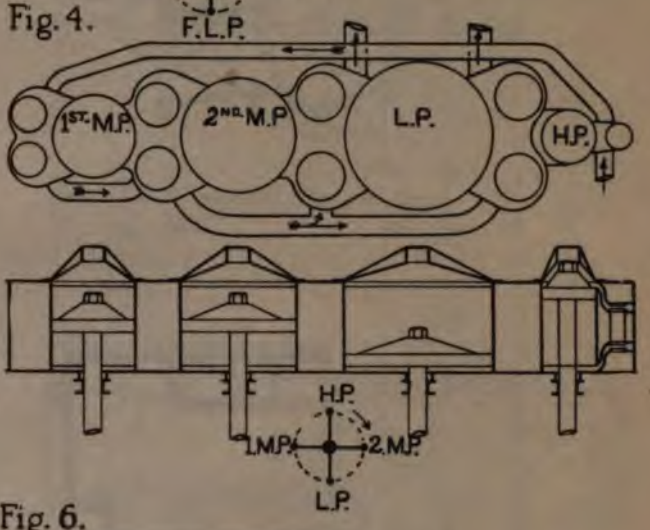

Fig. 5.

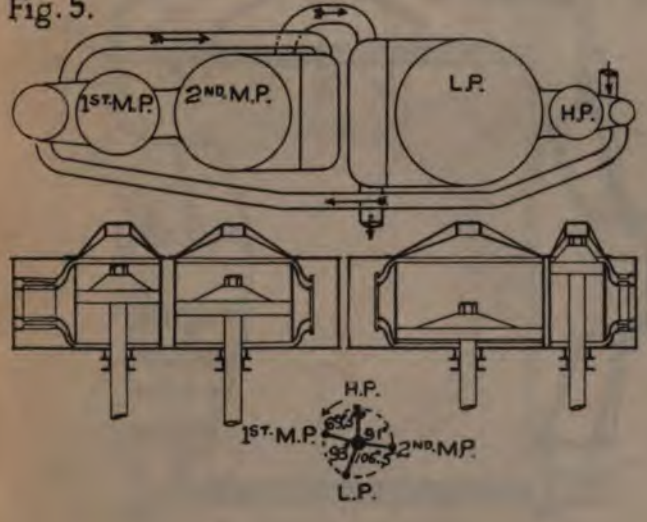

Fig. 6.

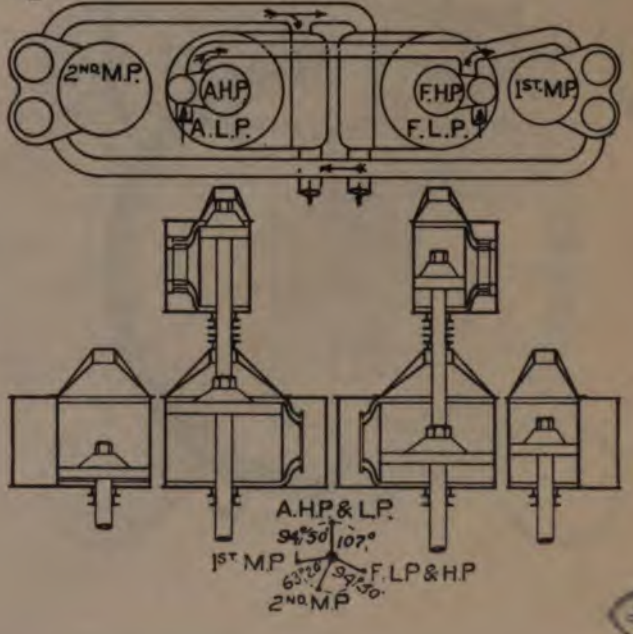

FIG. 1.
FIG. 2.

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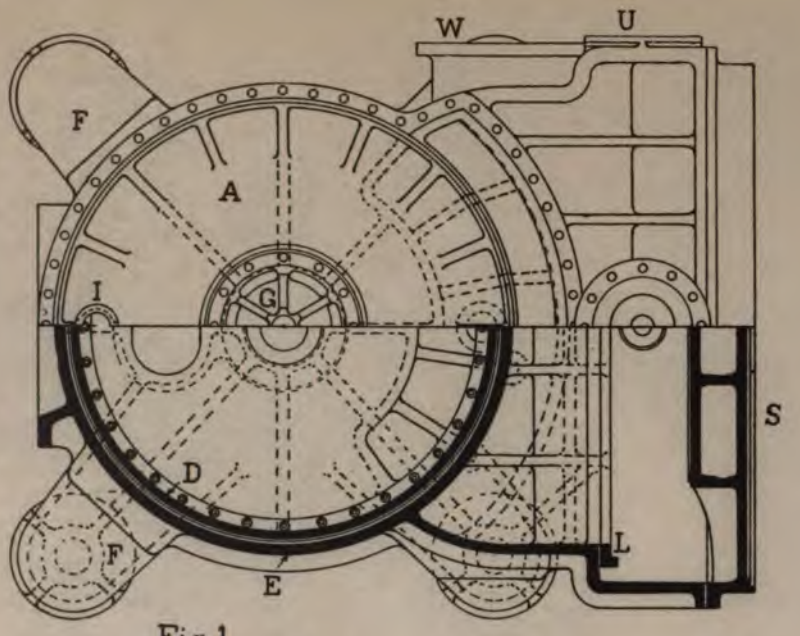


Fig.1.

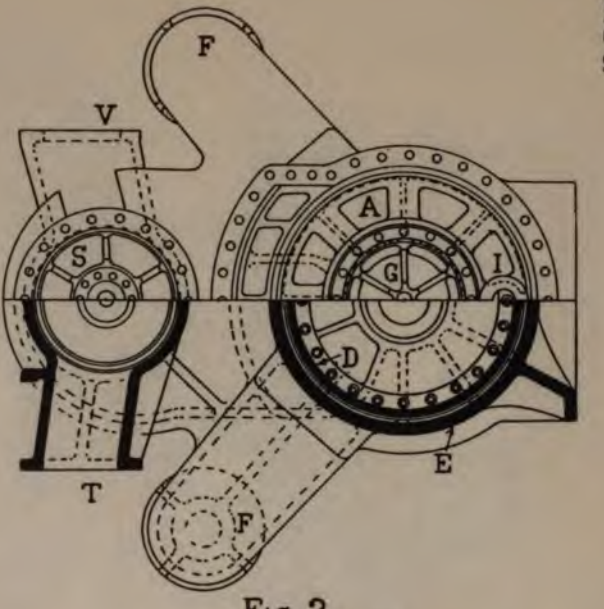
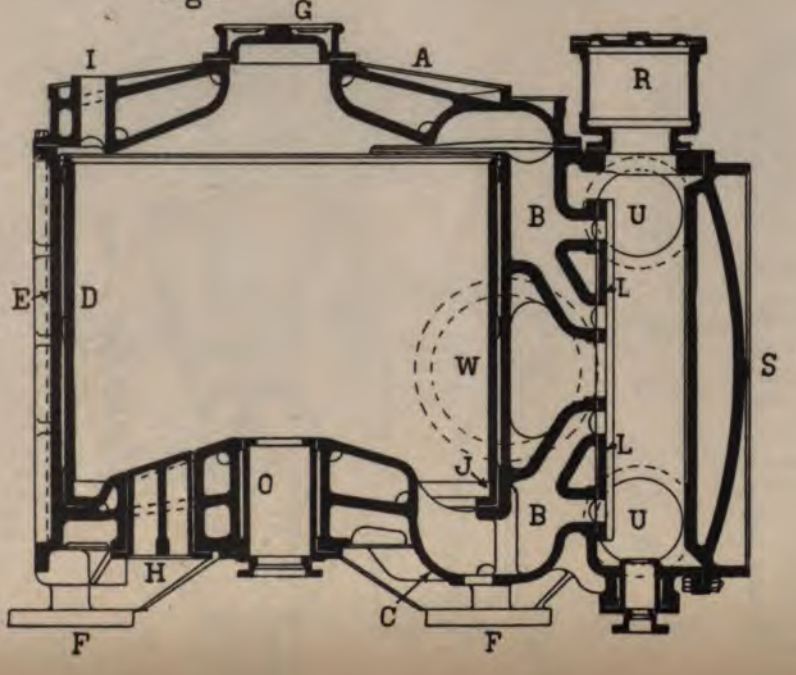
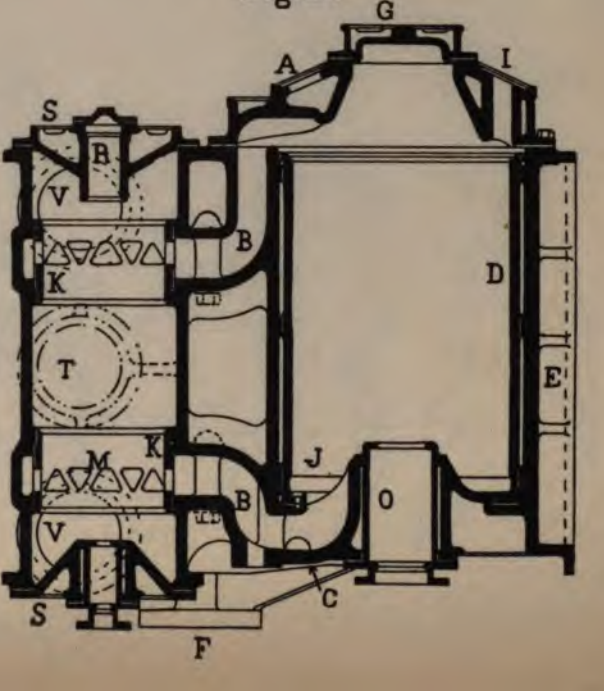
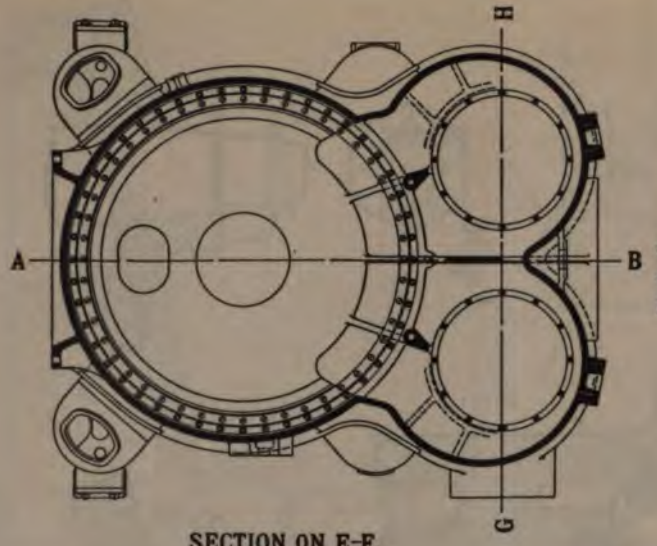


Fig.2.



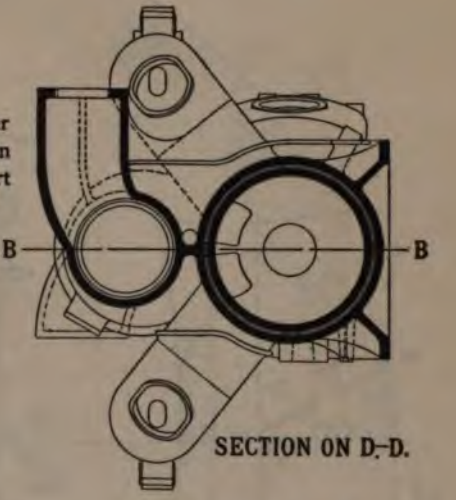




SECTION ON E.-F.

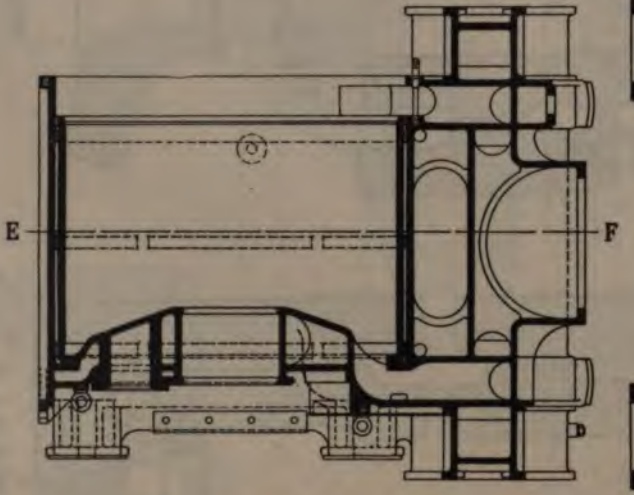
FIG. 2.

Observe the improvement in design over cylinders shown in Plate 9; particularly in shortening and straightening the steam port passages.

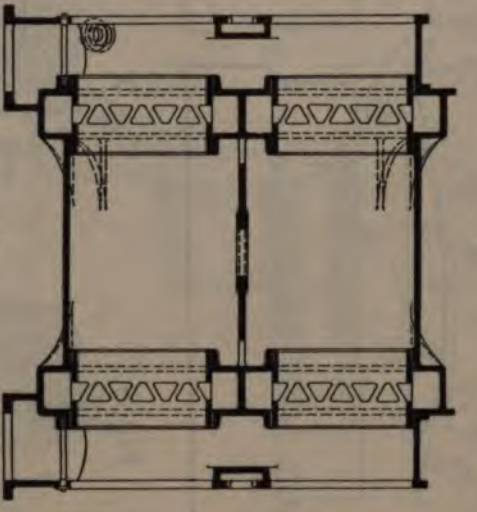


SECTION ON D.-D.

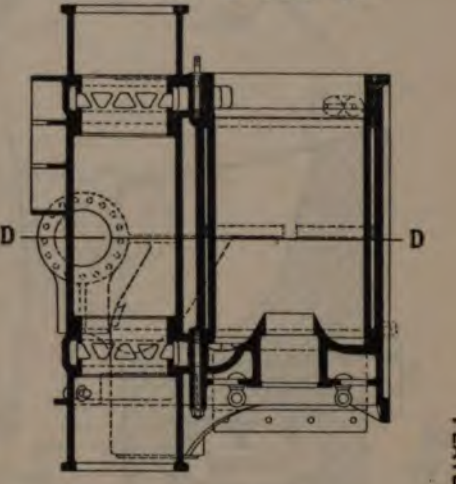
FIG. 1.



SECTION ON A.-B.



SECTION ON G.-H.



SECTION ON B.-B.

PLATE 10.

PLATE 7.

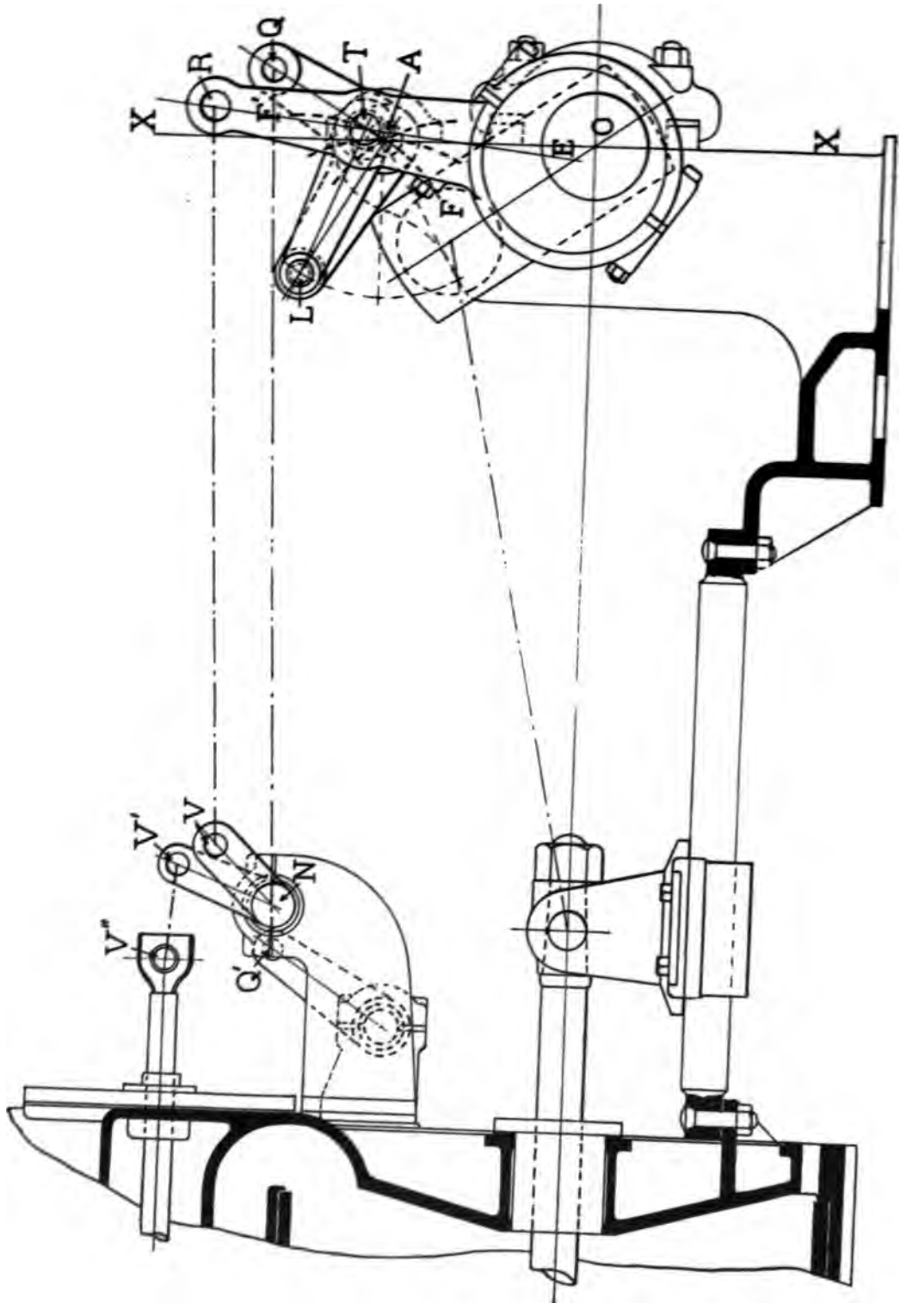


Fig. 1.

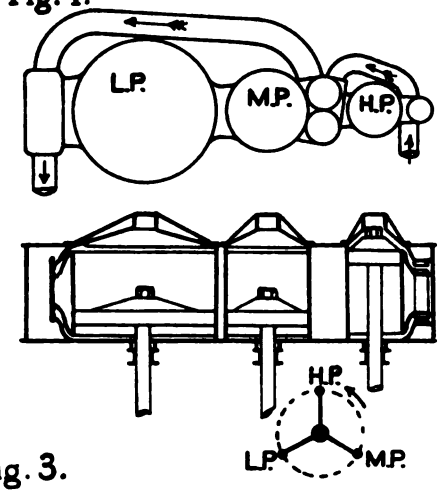


Fig. 2.

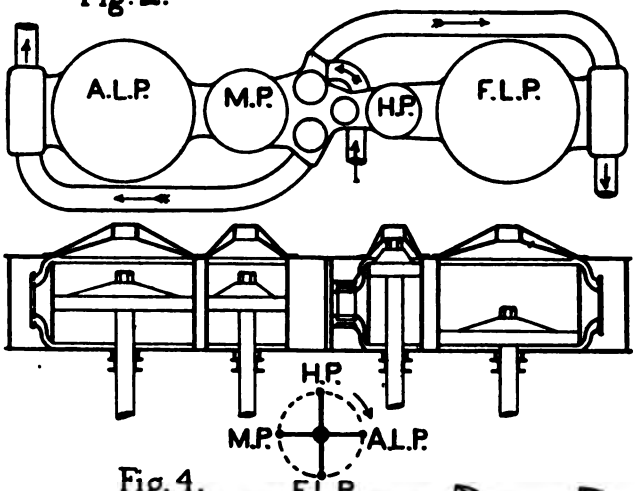


Fig. 3.

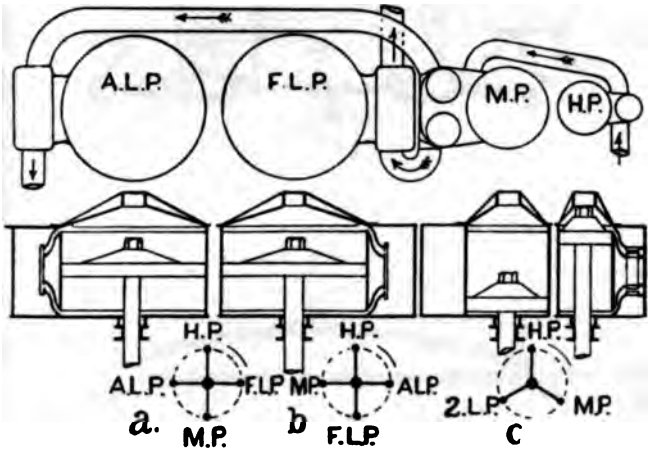


Fig. 4.

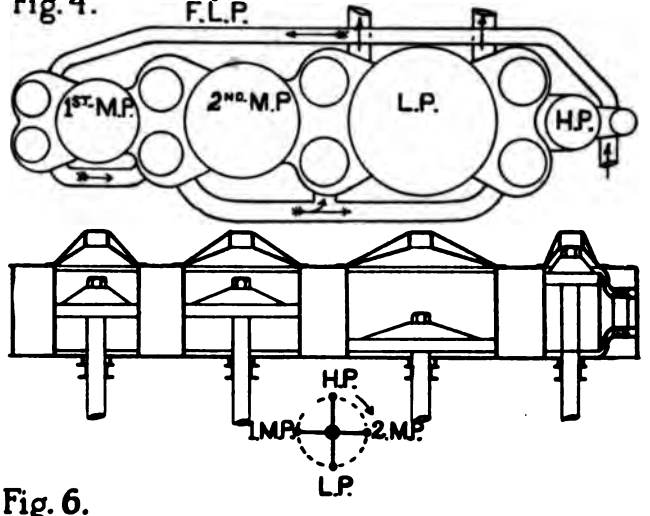


Fig. 5.

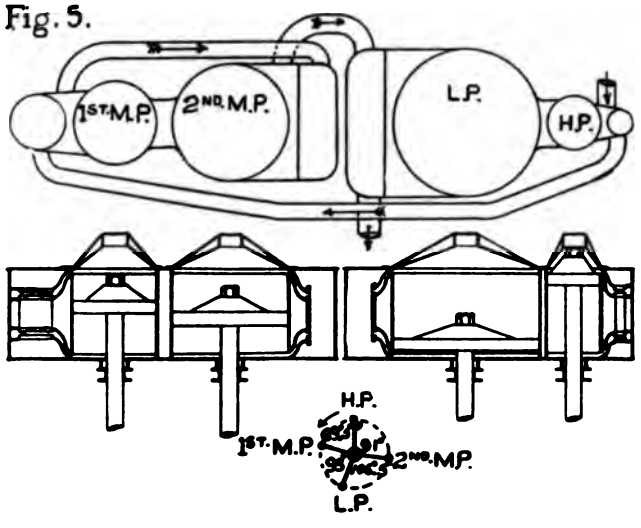


Fig. 6.

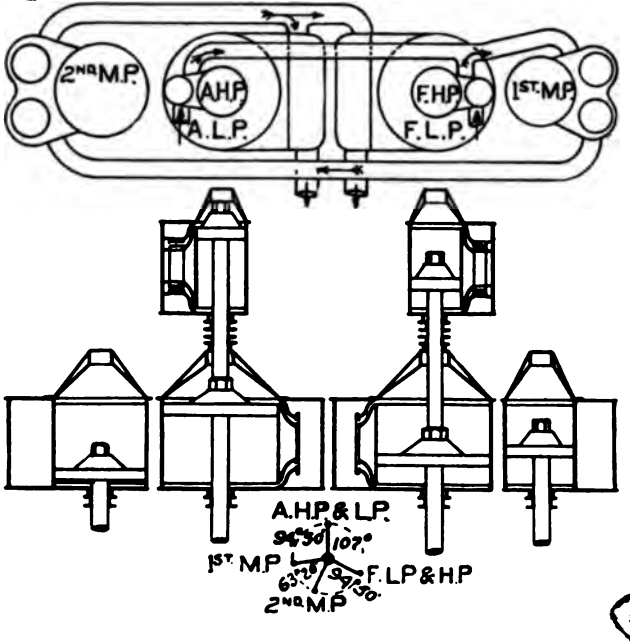


PLATE 11.

Fig.1.

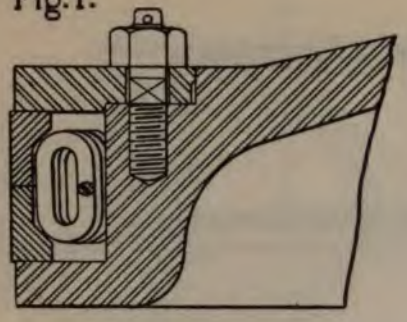


Fig.2.

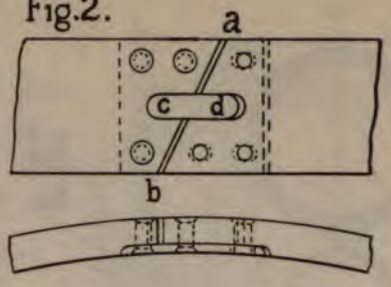
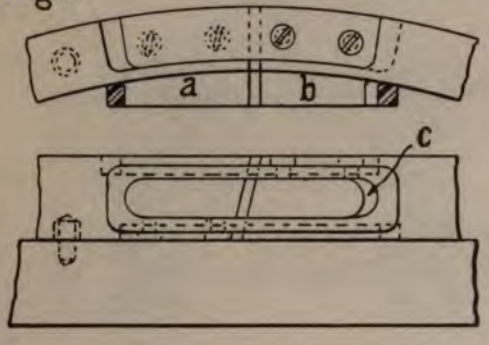


Fig.3.



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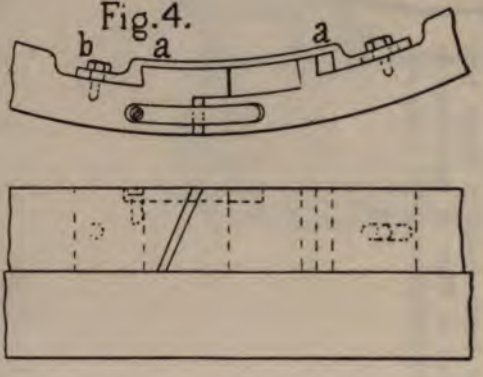


Fig.6.

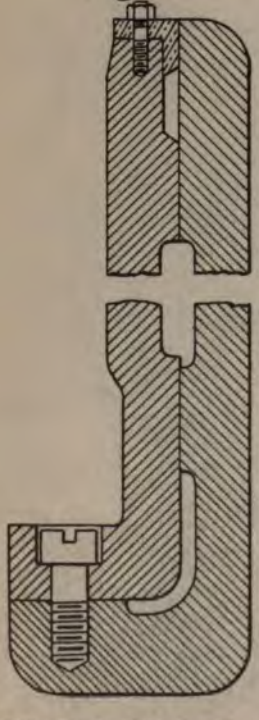


Fig.7.

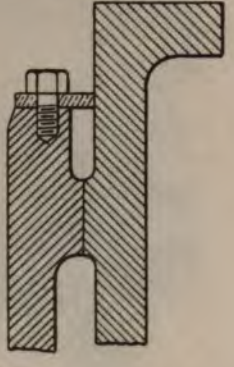


Fig.8.

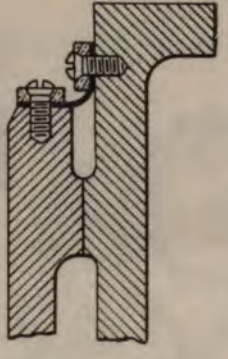
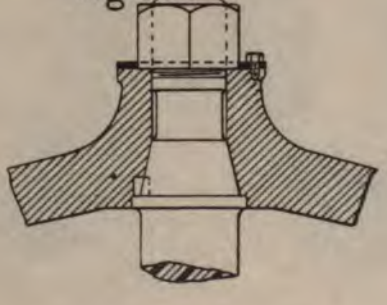
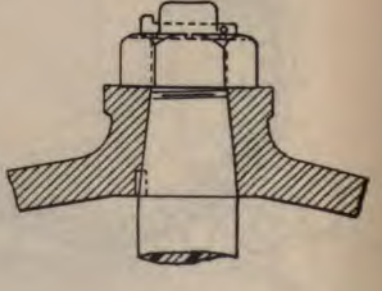


Fig.5.

Fig.10.

Fig.9.





This Method is Preferred.

Fig. 1.

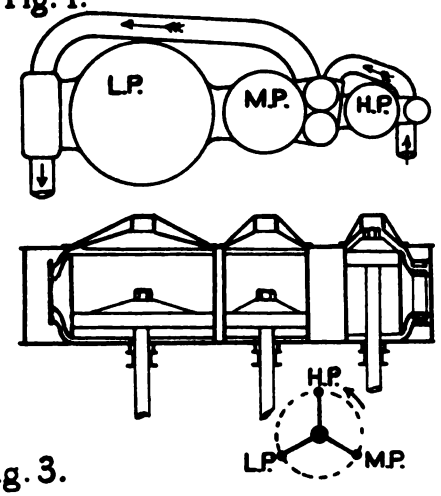


Fig. 2.

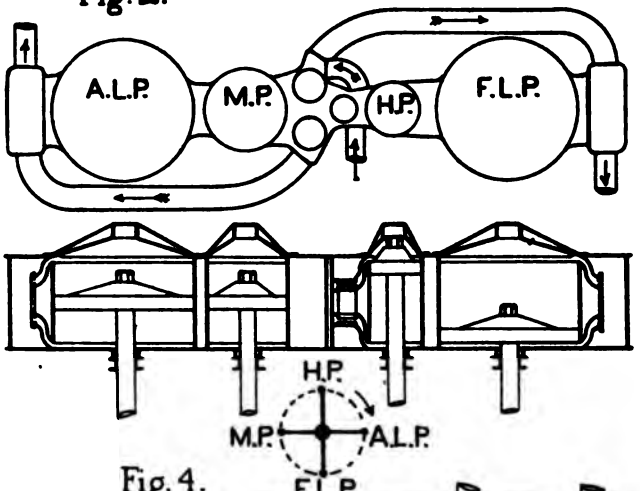


Fig. 3.

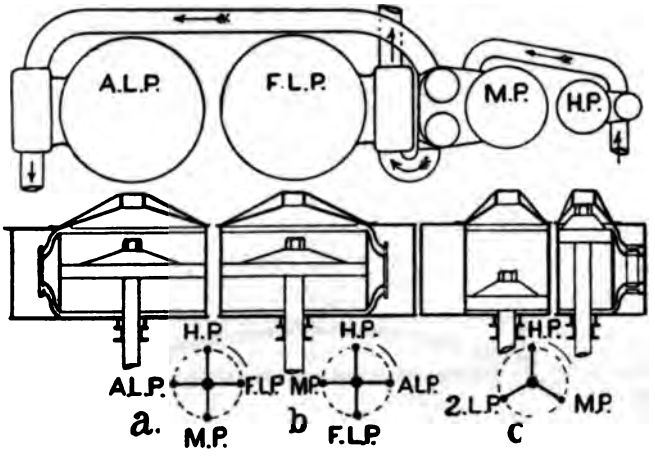
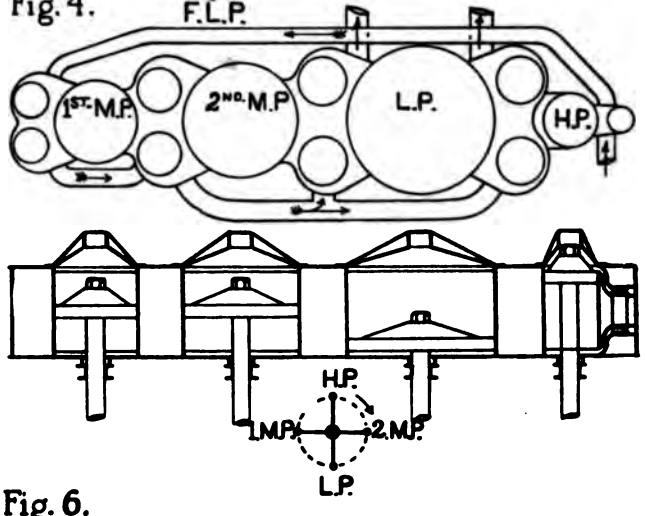


Fig. 4.



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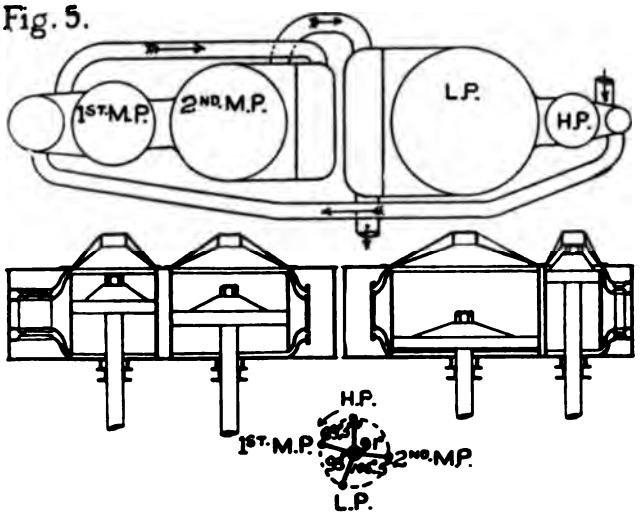

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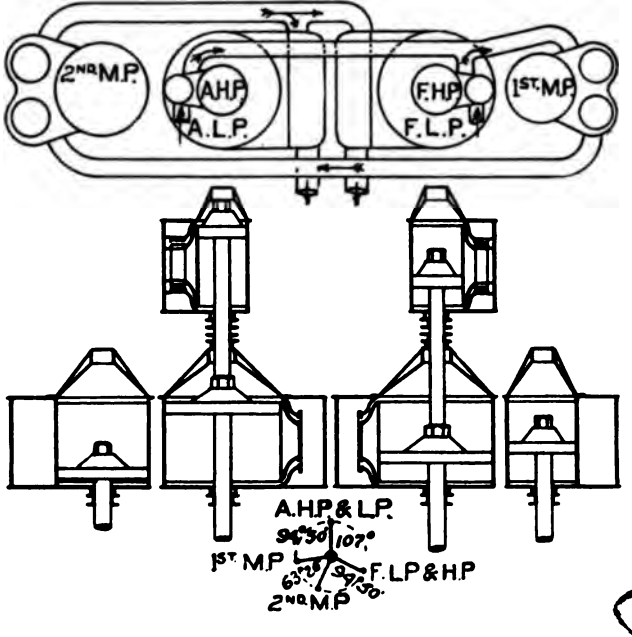
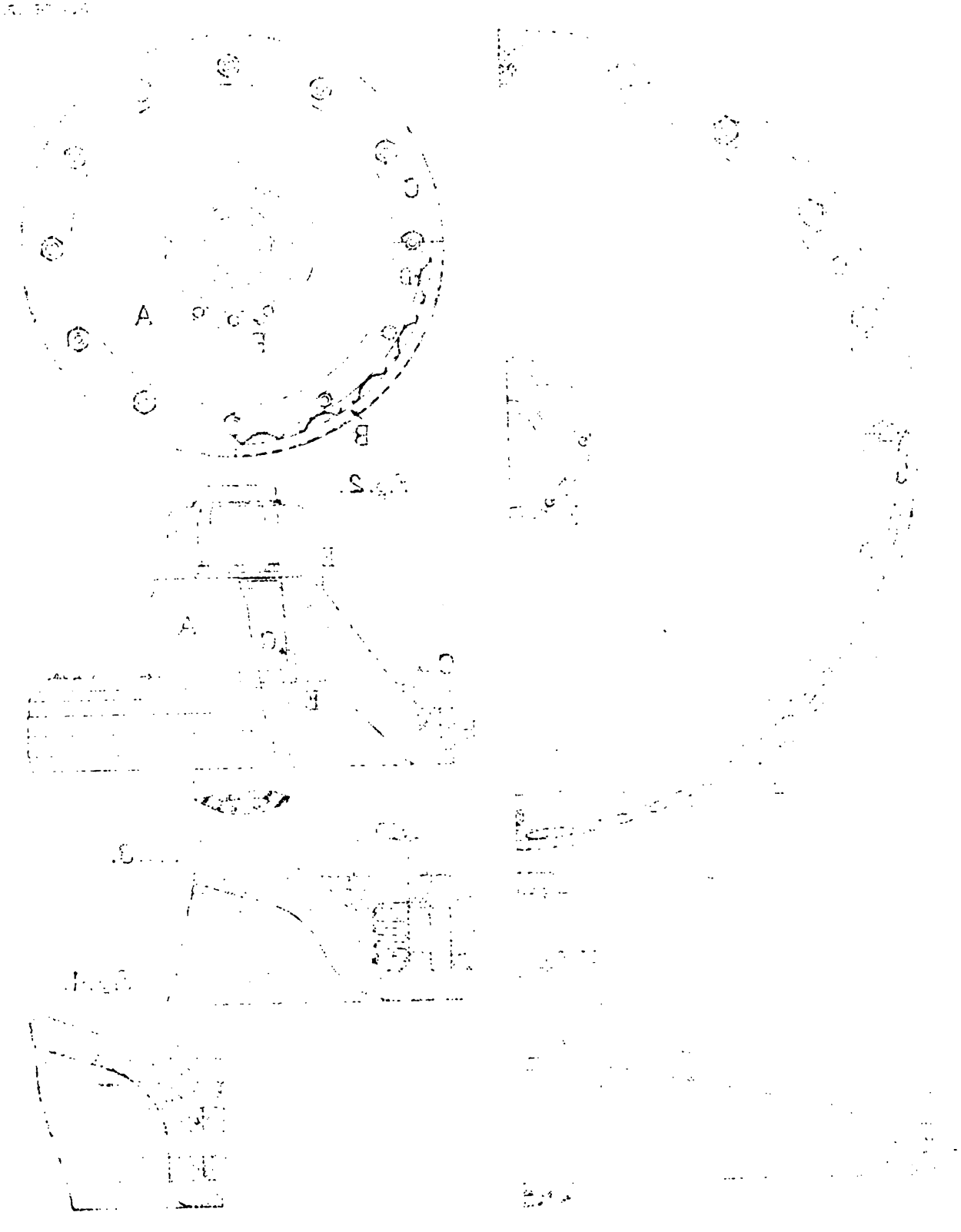



A
B

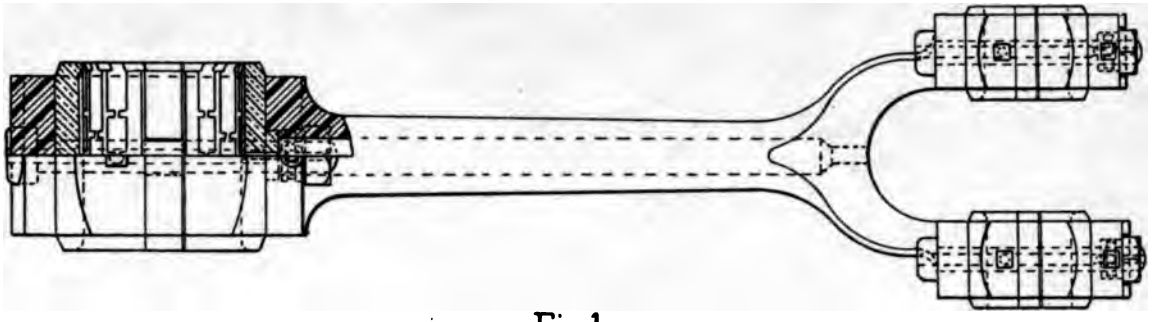


Fig.1.

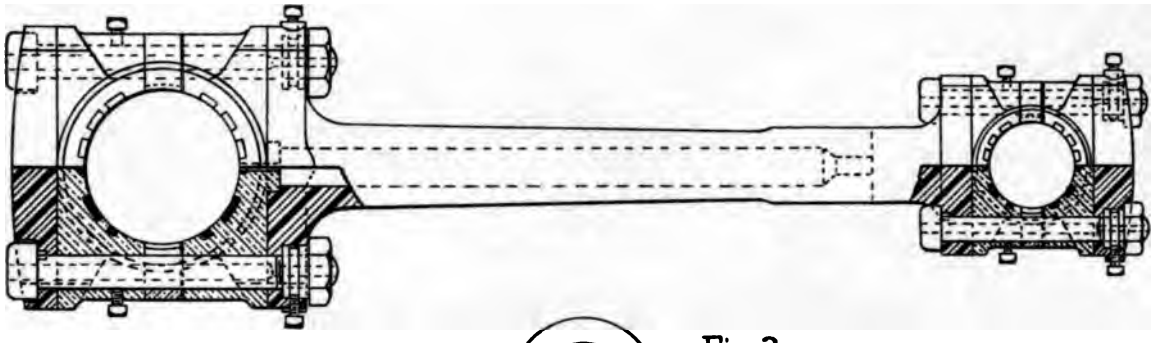
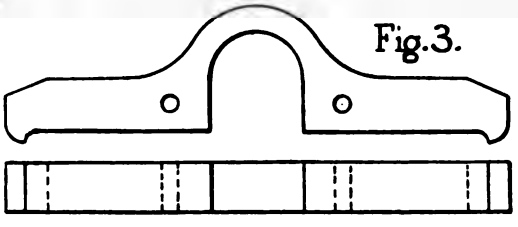
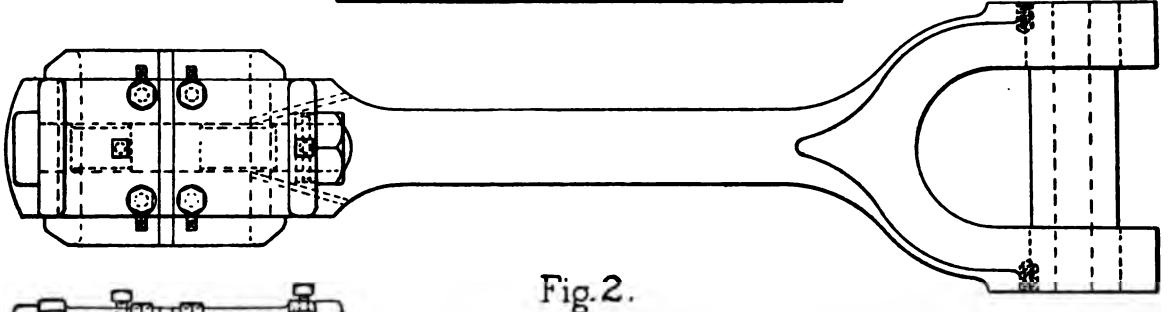


Fig.3.





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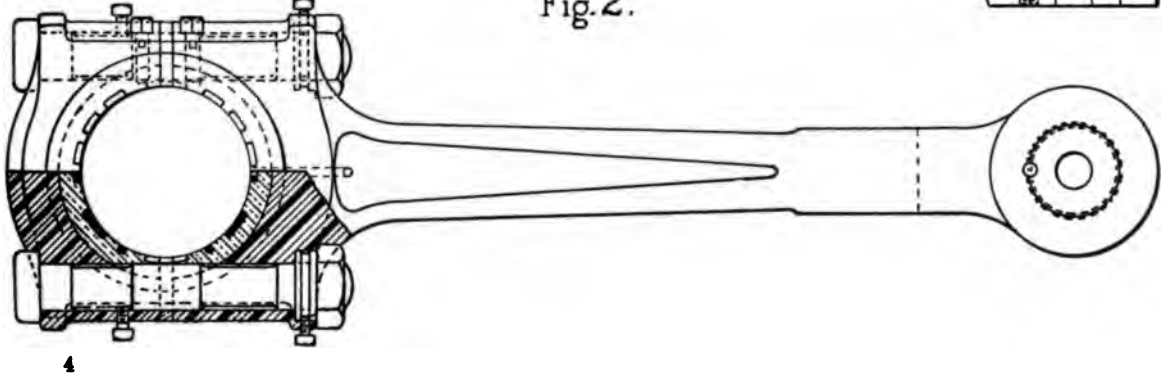


PLATE 9.

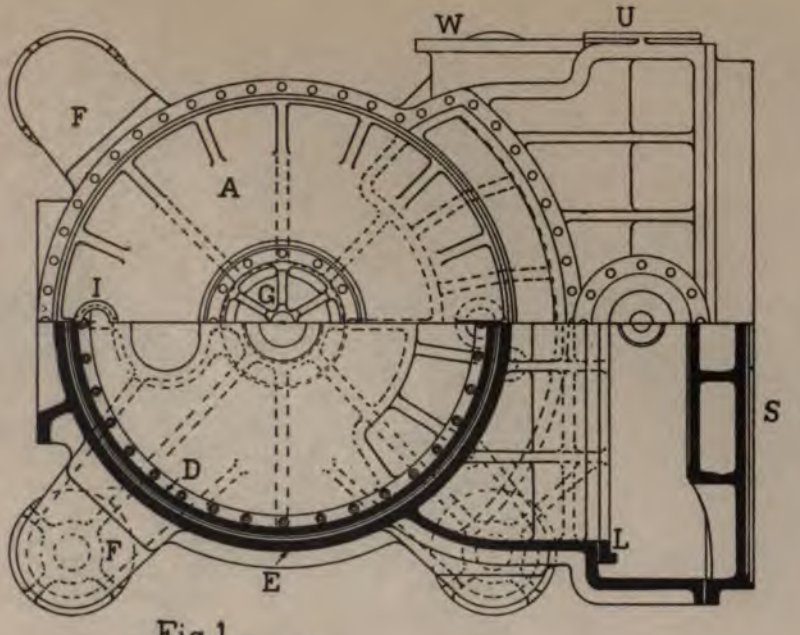
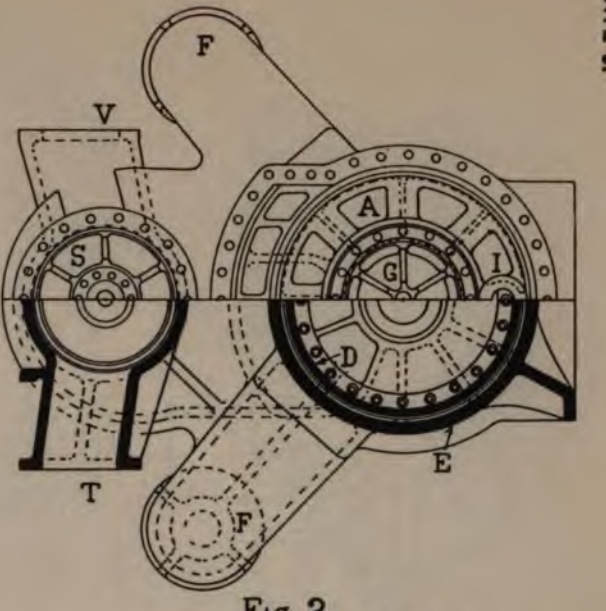


Fig.1.



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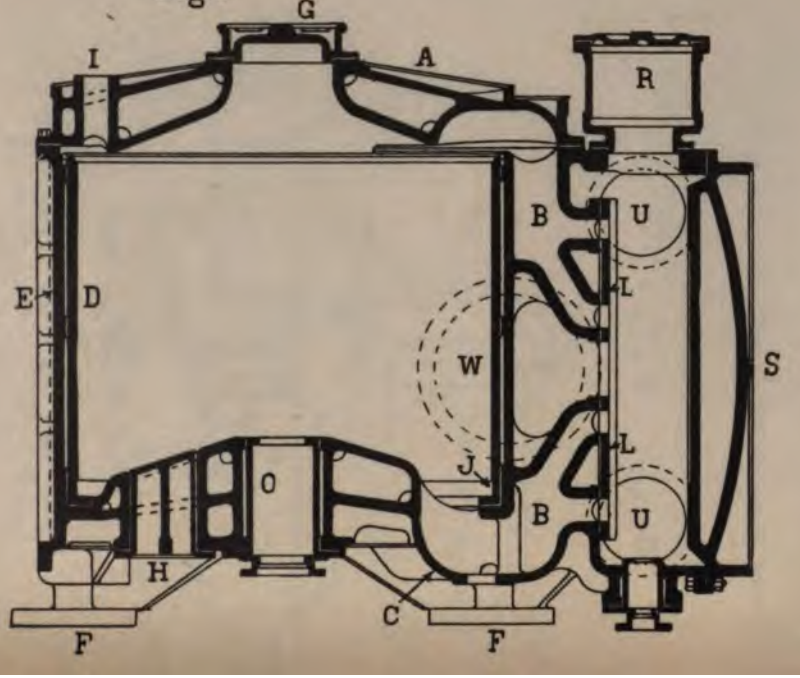
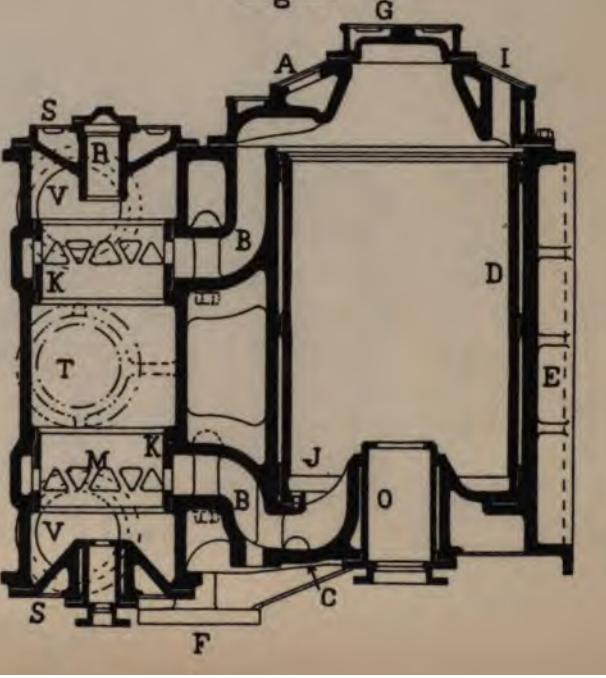
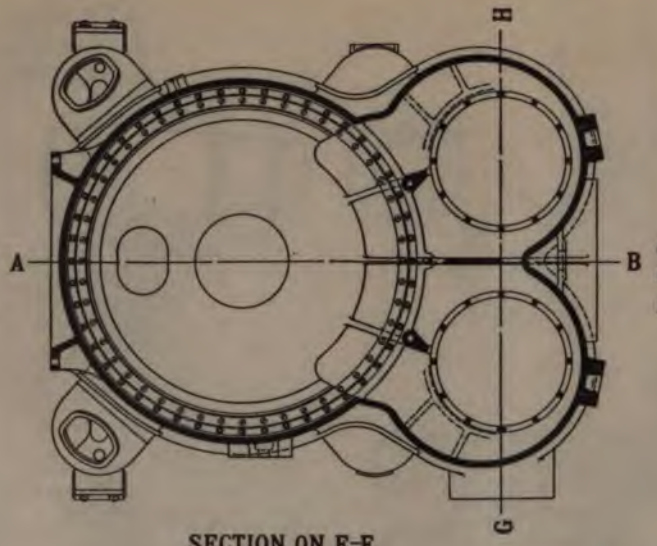


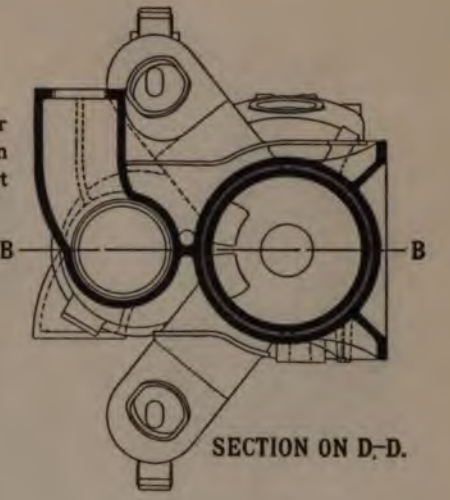




SECTION ON E.-F.

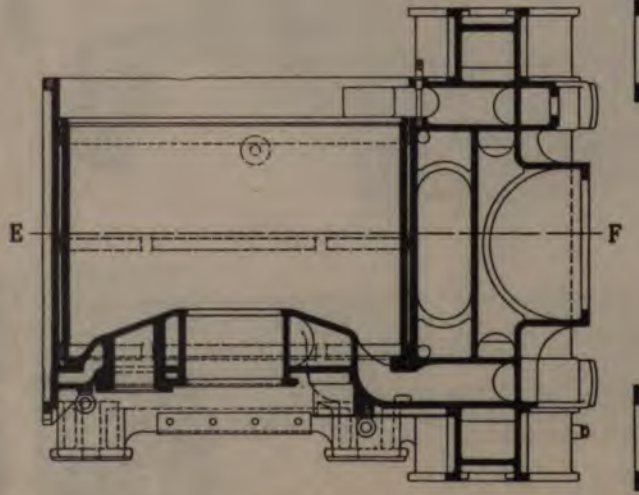
FIG. 2.

Observe the improvement in design over cylinders shown in Plate 9; particularly in shortening and straightening the steam port passages.

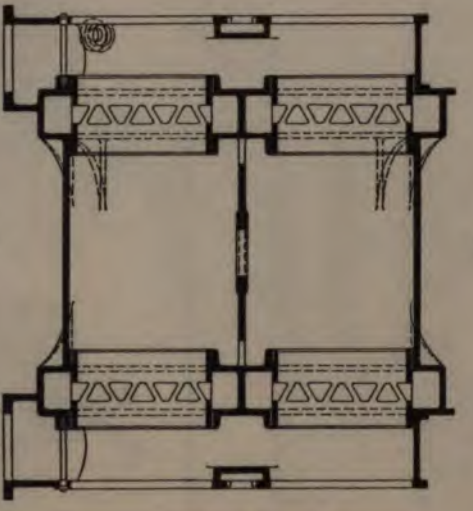


SECTION ON D.-D.

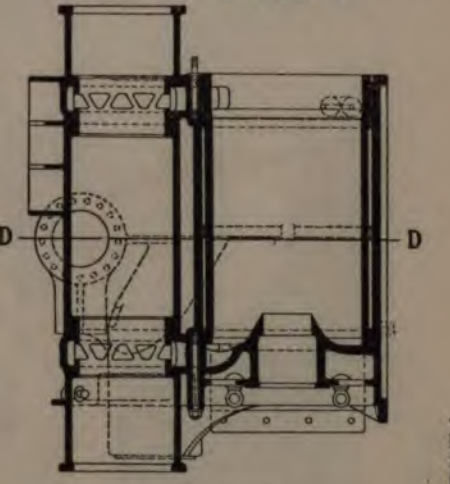
FIG. 1.



SECTION ON A.-B.



SECTION ON G.-H.



SECTION ON B.-B.

PLATE 10.

SECTION ON B-B
SECTION ON C-C
SECTION ON A-A
SECTION ON D-D
SECTION ON E-E

FIG. 1.

FIG. 2.

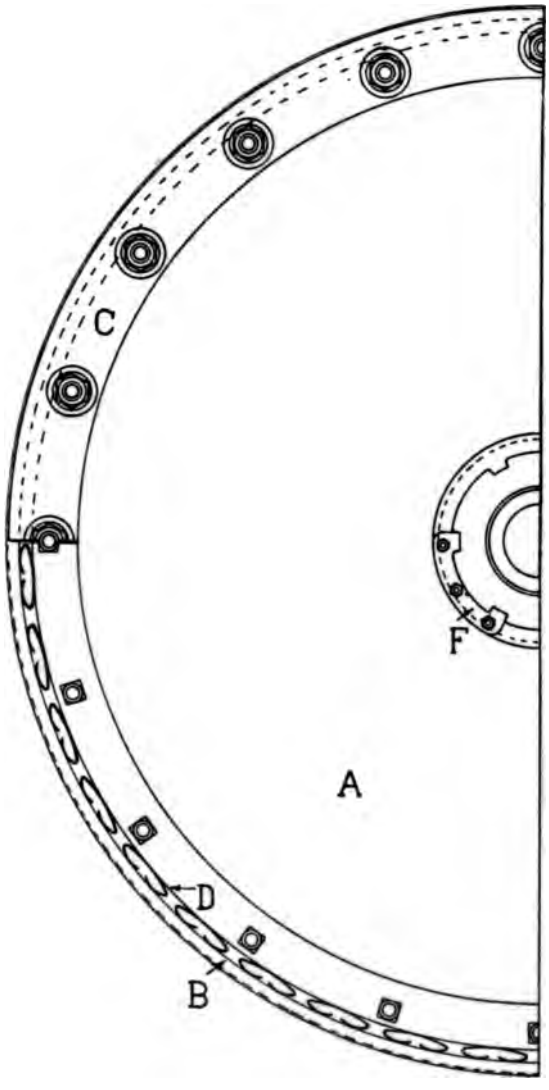


Fig.1.

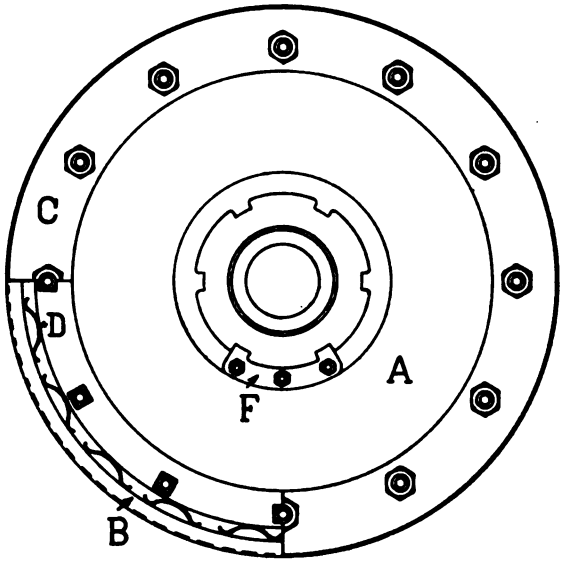
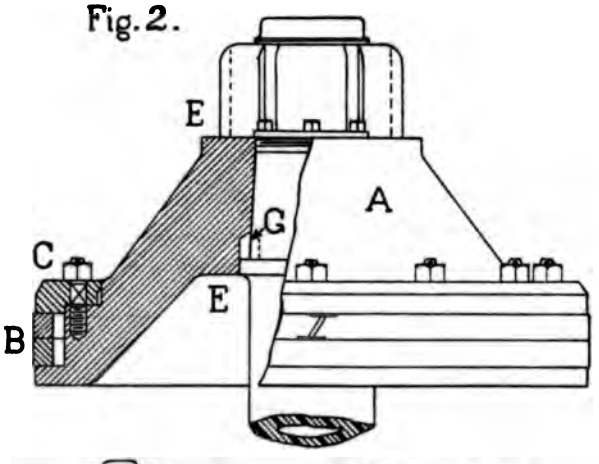
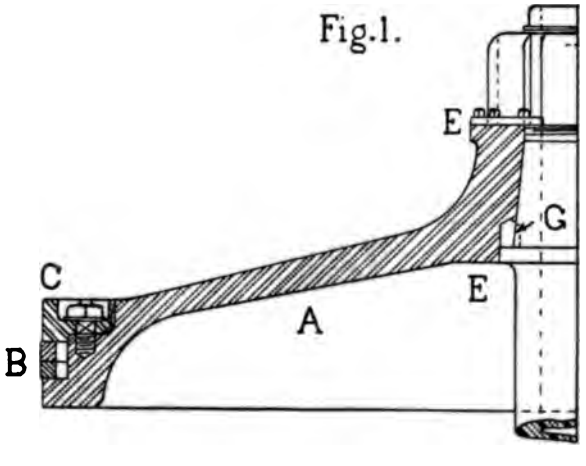


Fig.2.





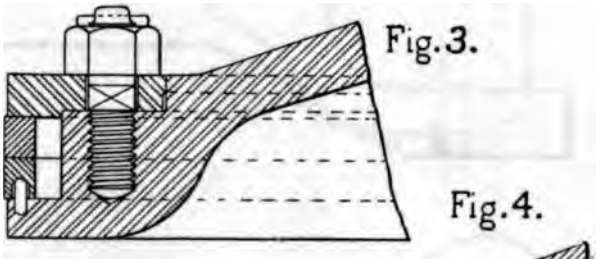


Fig.3.

Fig.4.

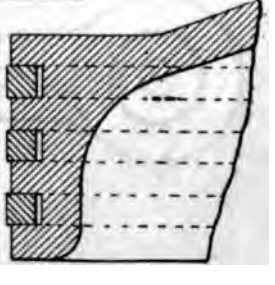


PLATE 11.

Fig.1.

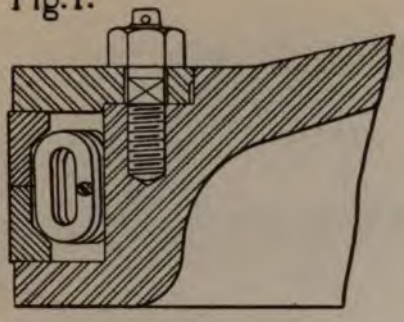
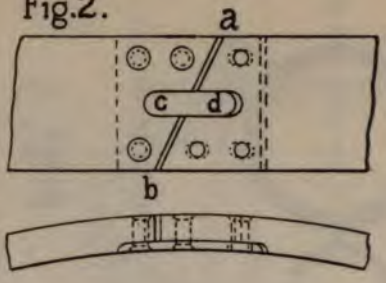


Fig.2.



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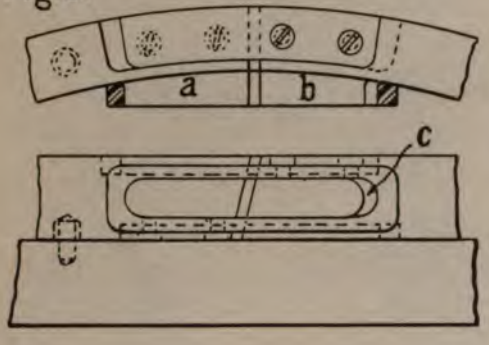


Fig.4.

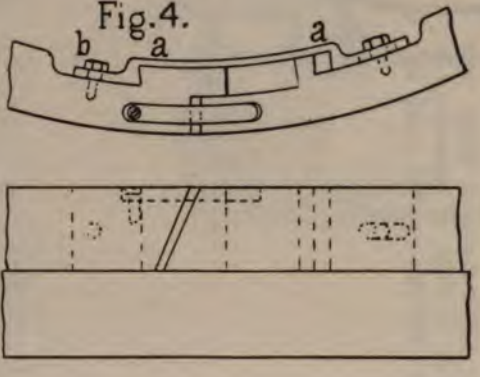


Fig.6.

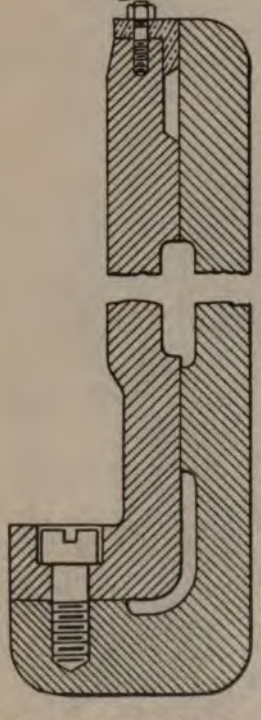


Fig.7.

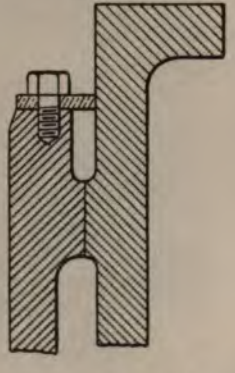


Fig.8.

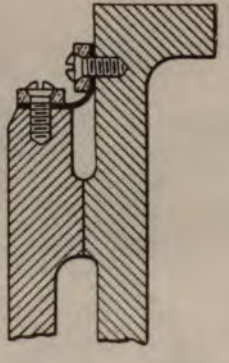


Fig.5.

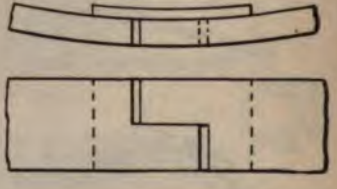
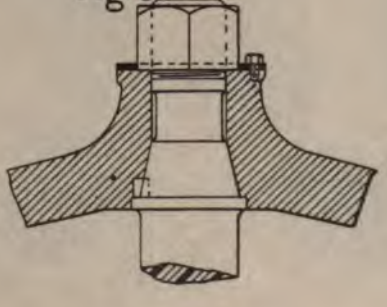
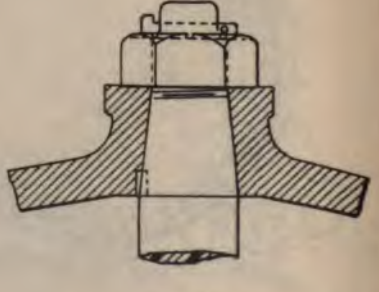


Fig.10.

Fig.9.





This Method is Preferred.

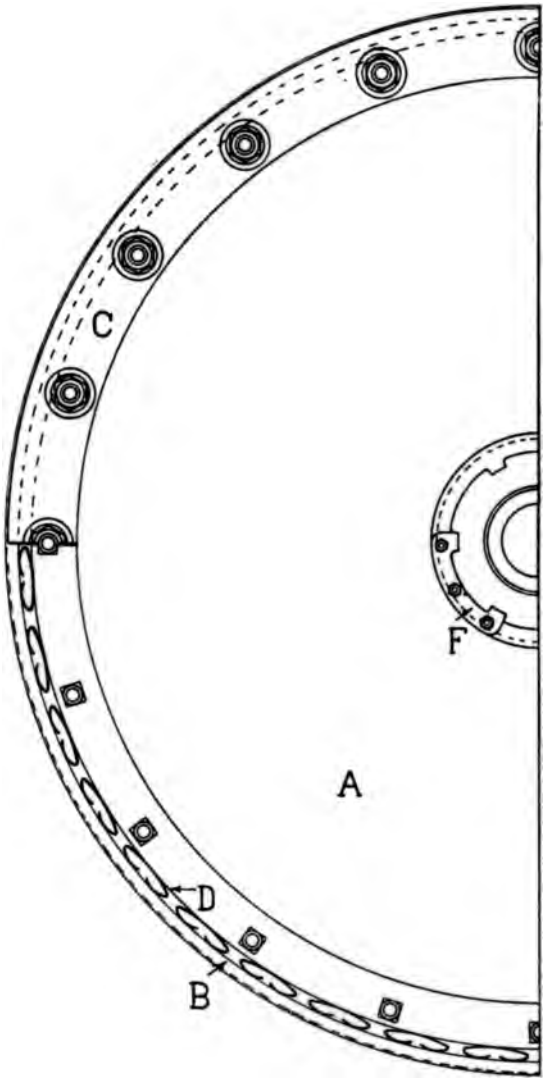
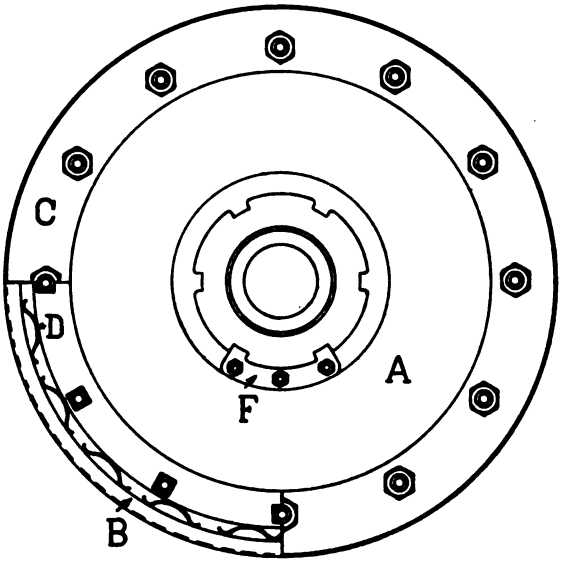


Fig.1.



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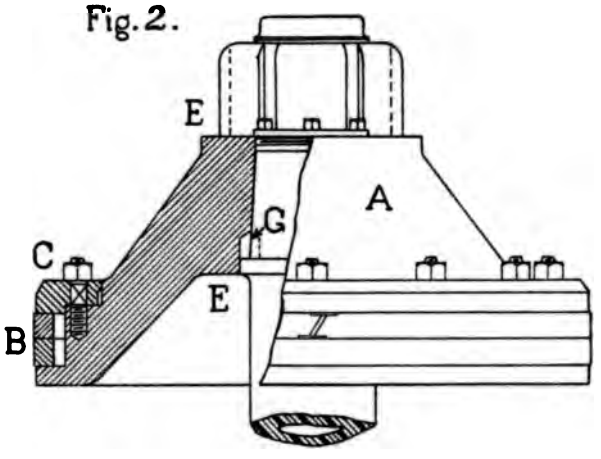


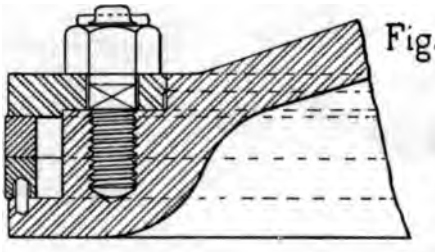
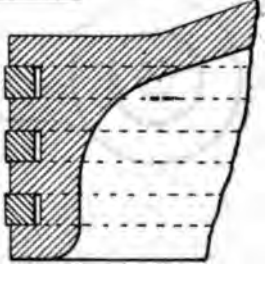
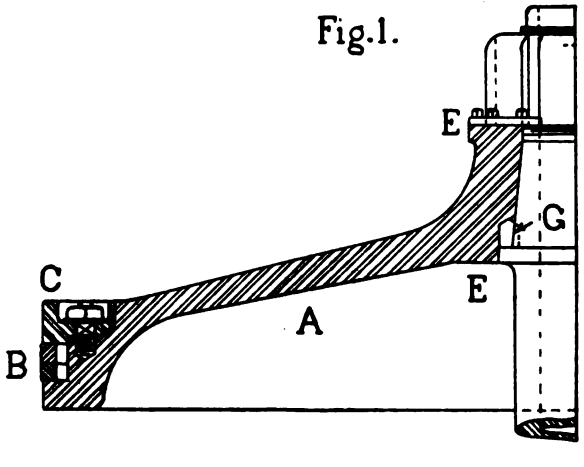
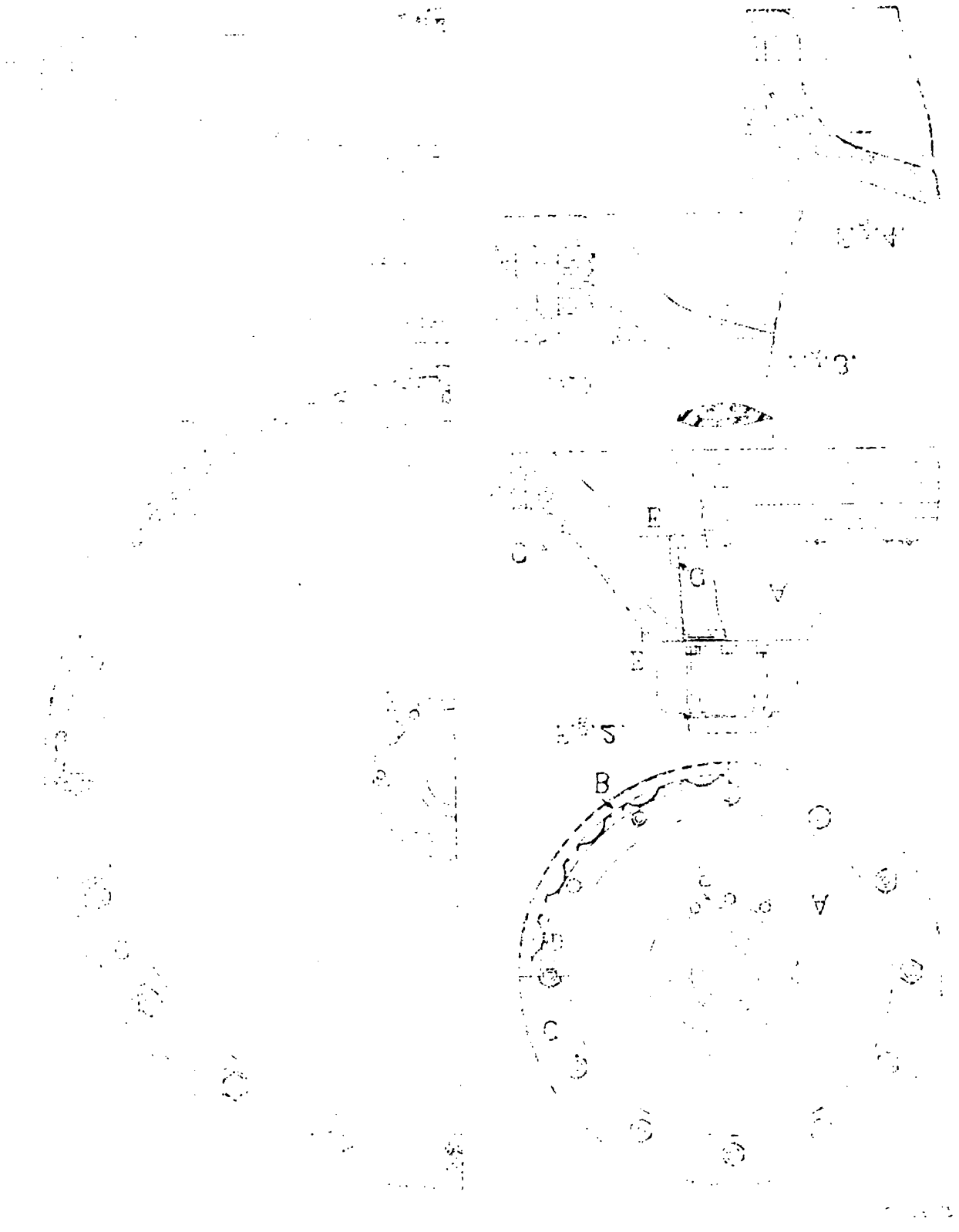


Fig.3.

Fig.4.



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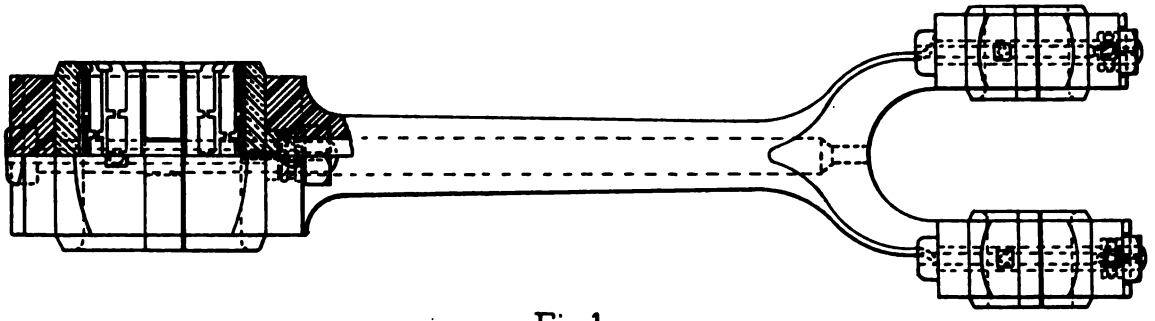


Fig.1.

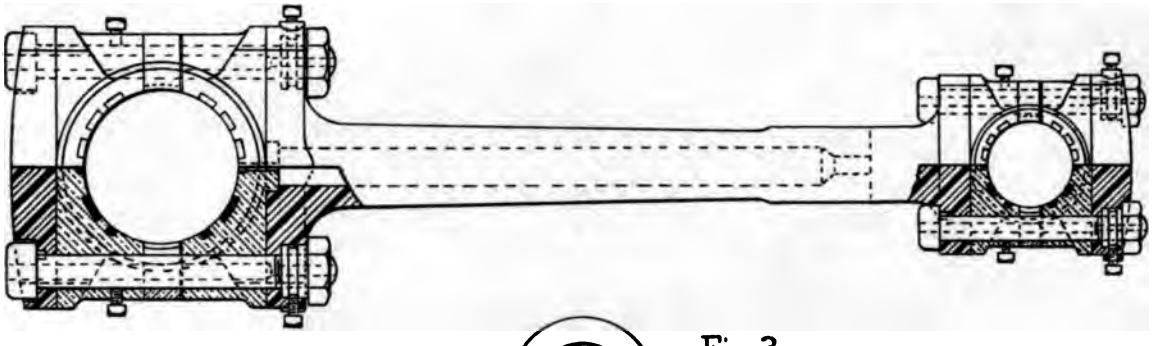
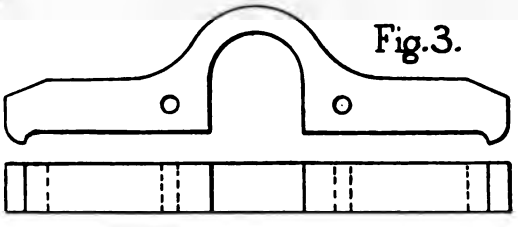


Fig.3.



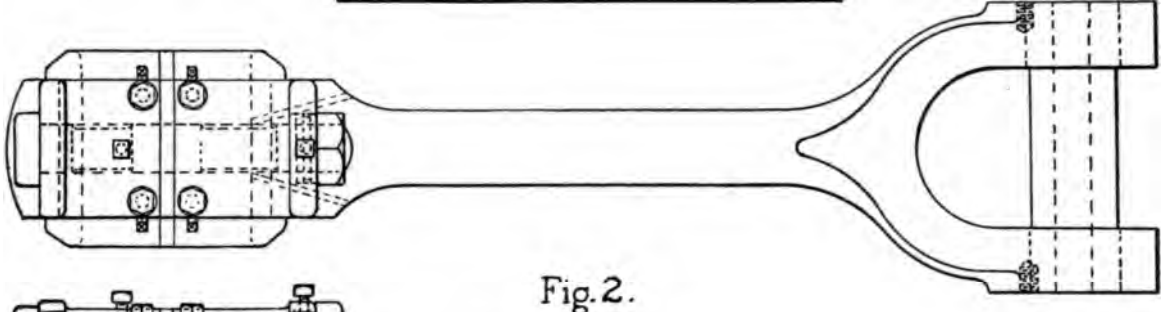


Fig.2.

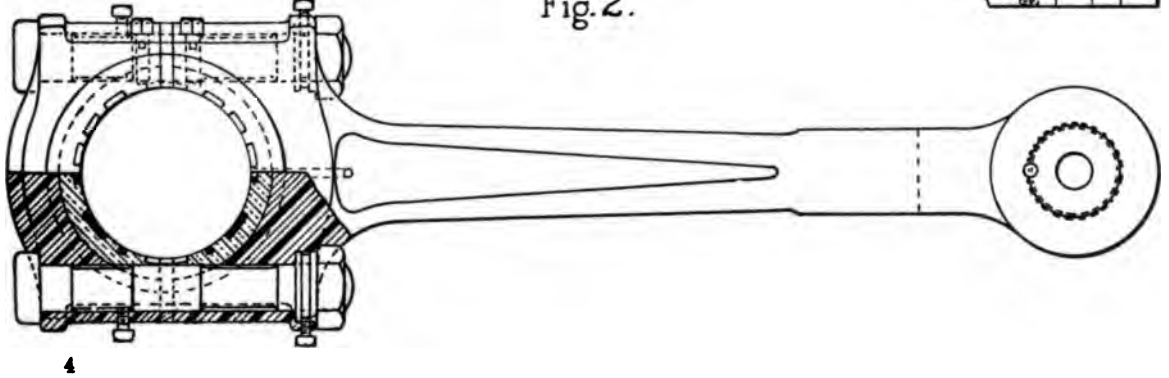


PLATE 14.

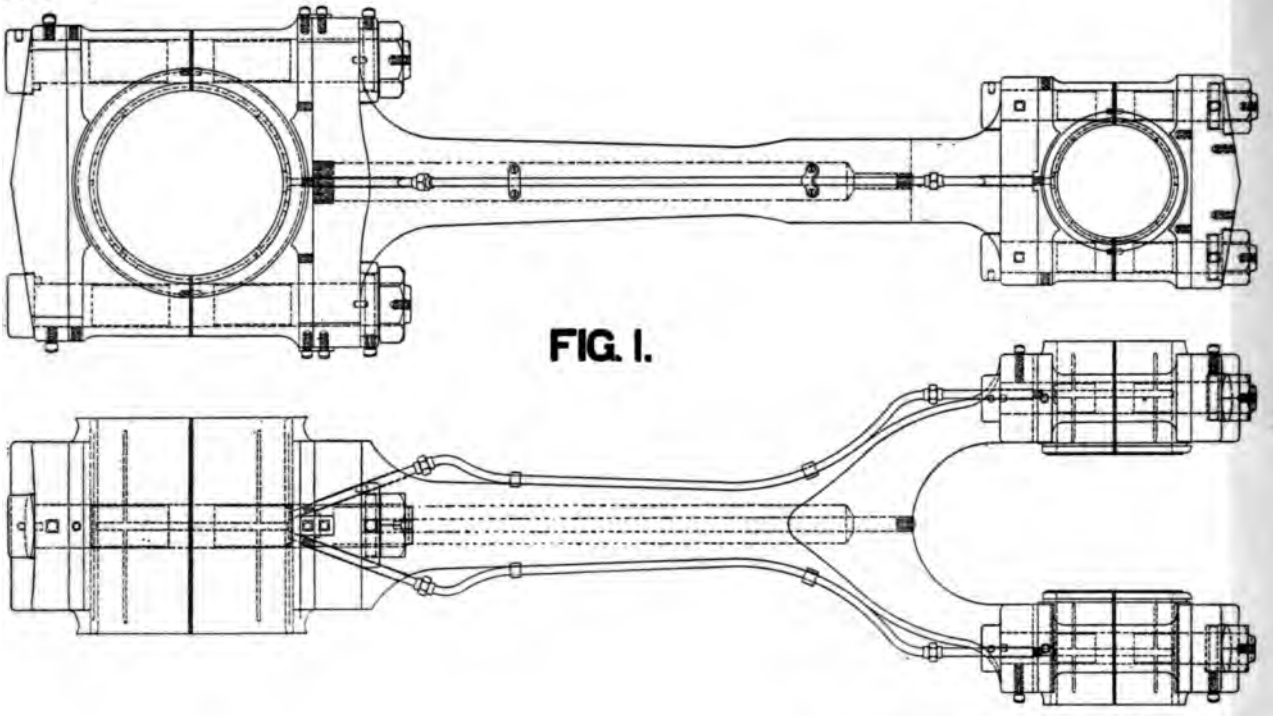


FIG. I.

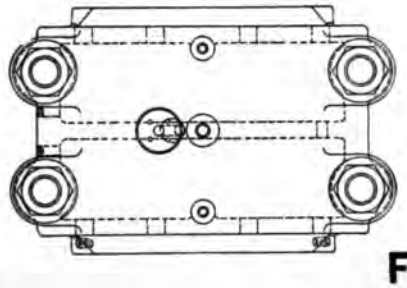
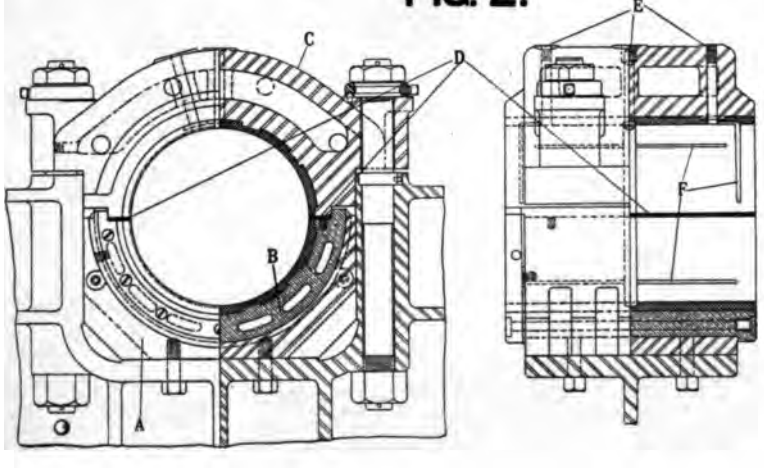


FIG. 2.



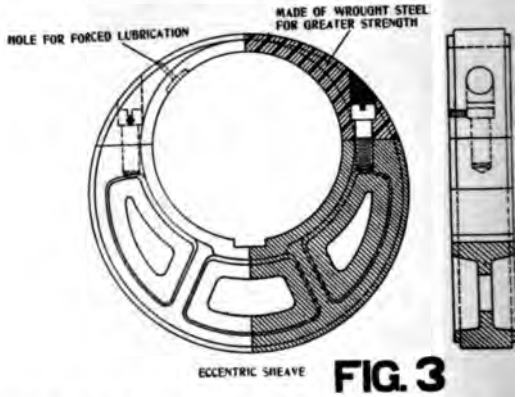


FIG. 3

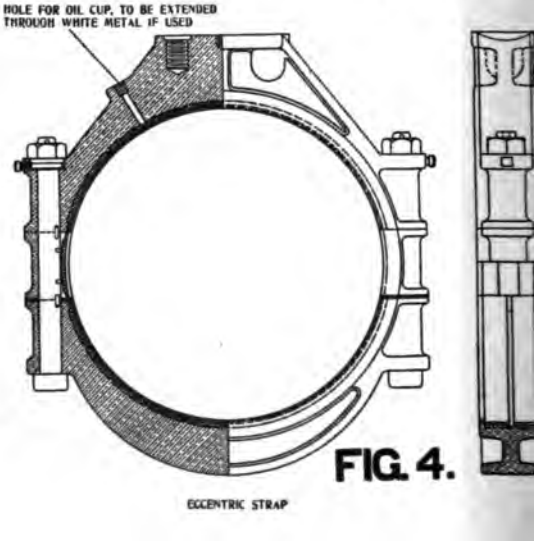


FIG. 4.

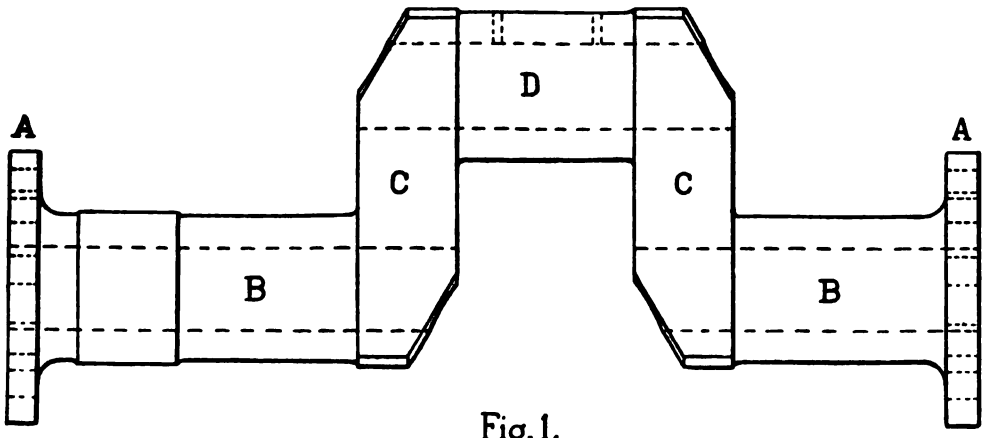
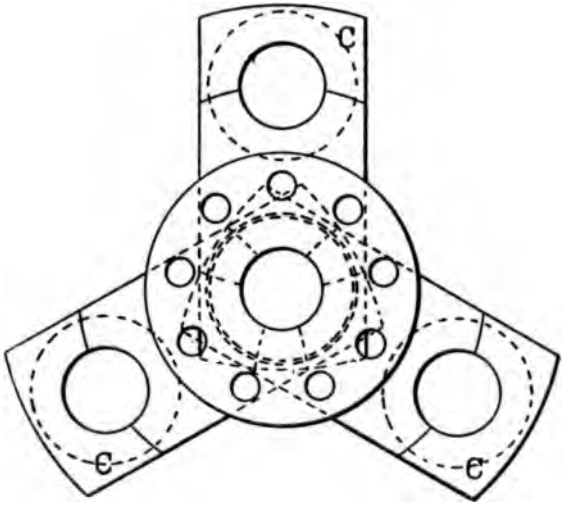
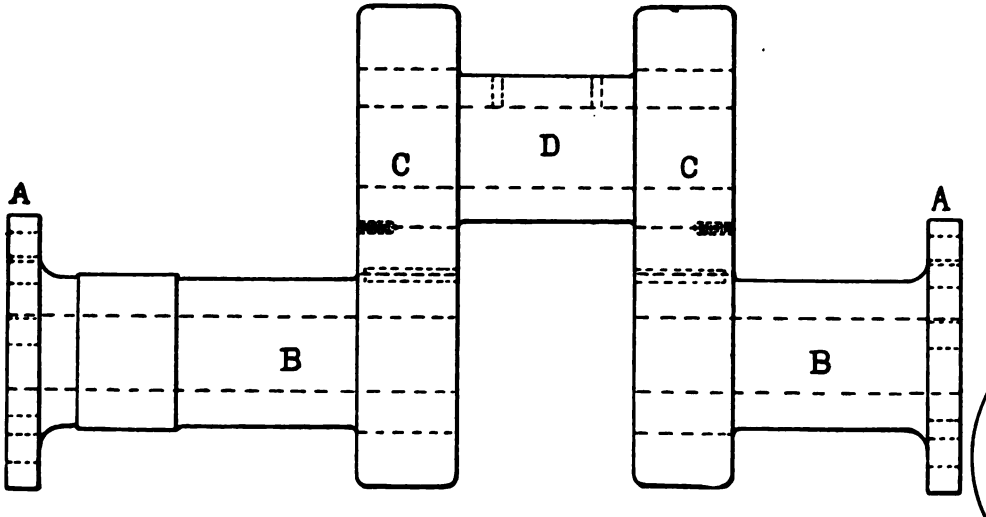


Fig. 1.





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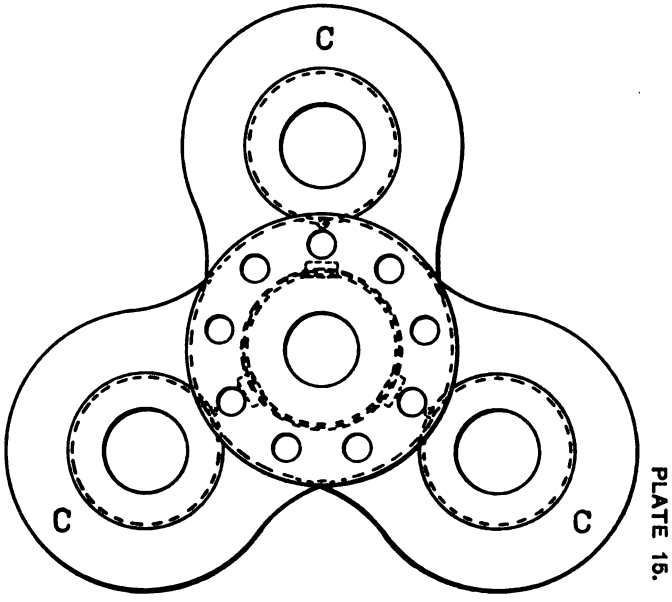


PLATE 15.

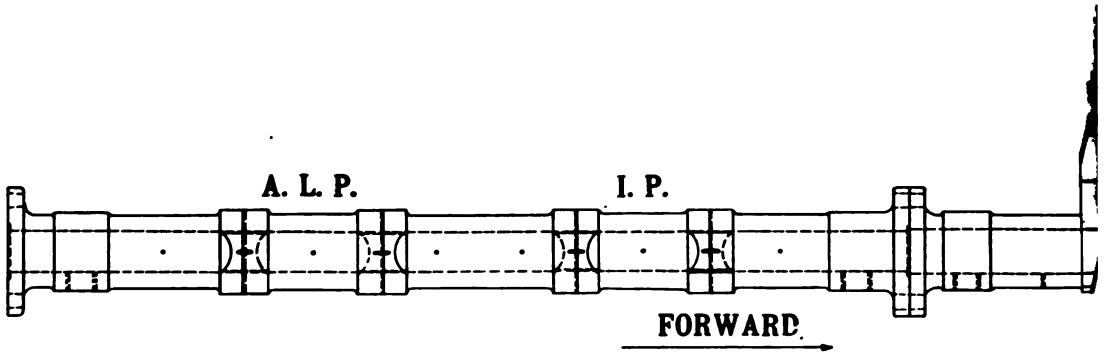
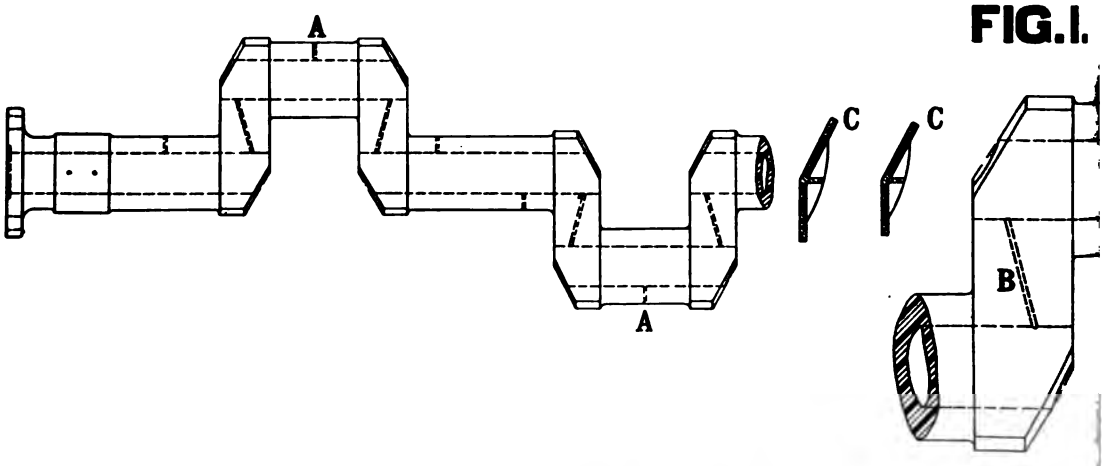
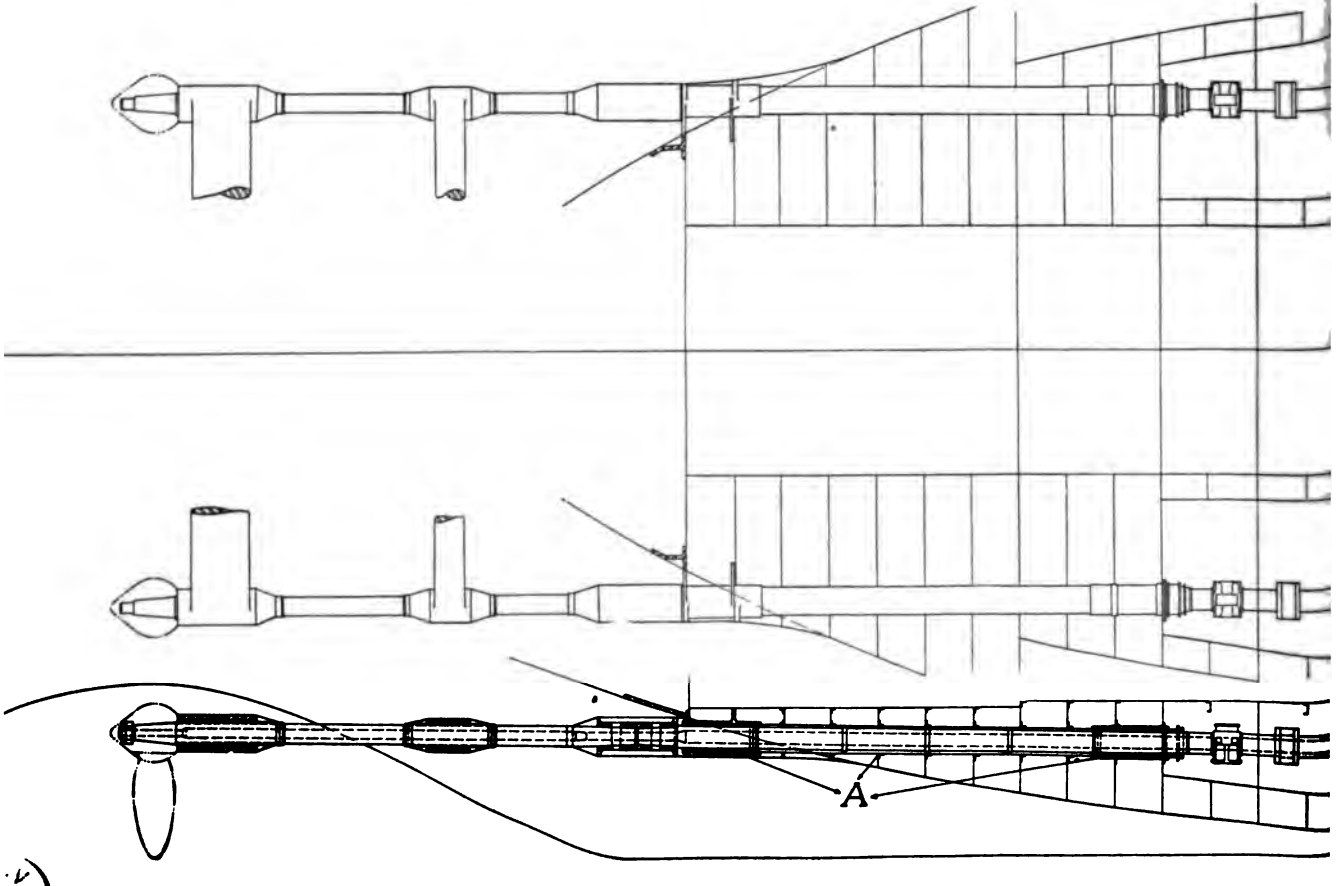
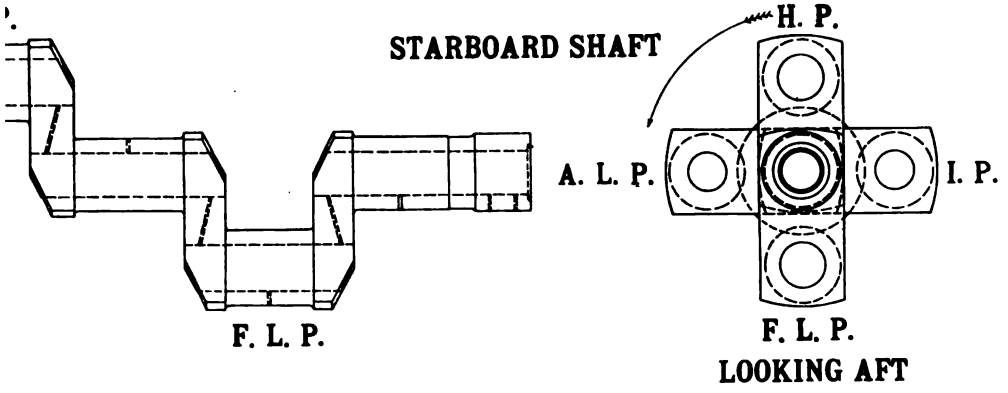
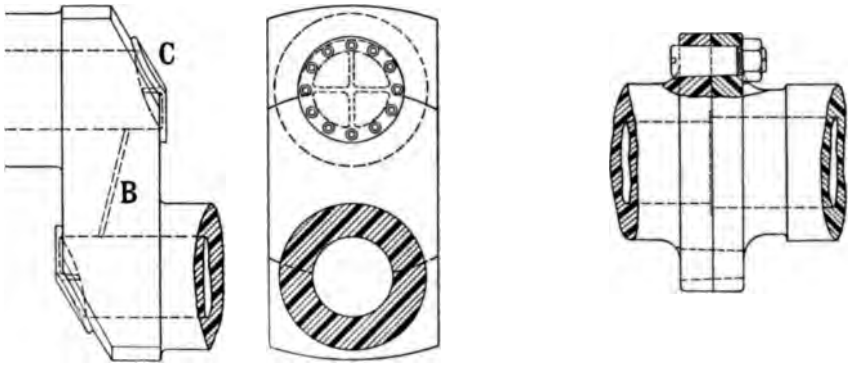


FIG. I.









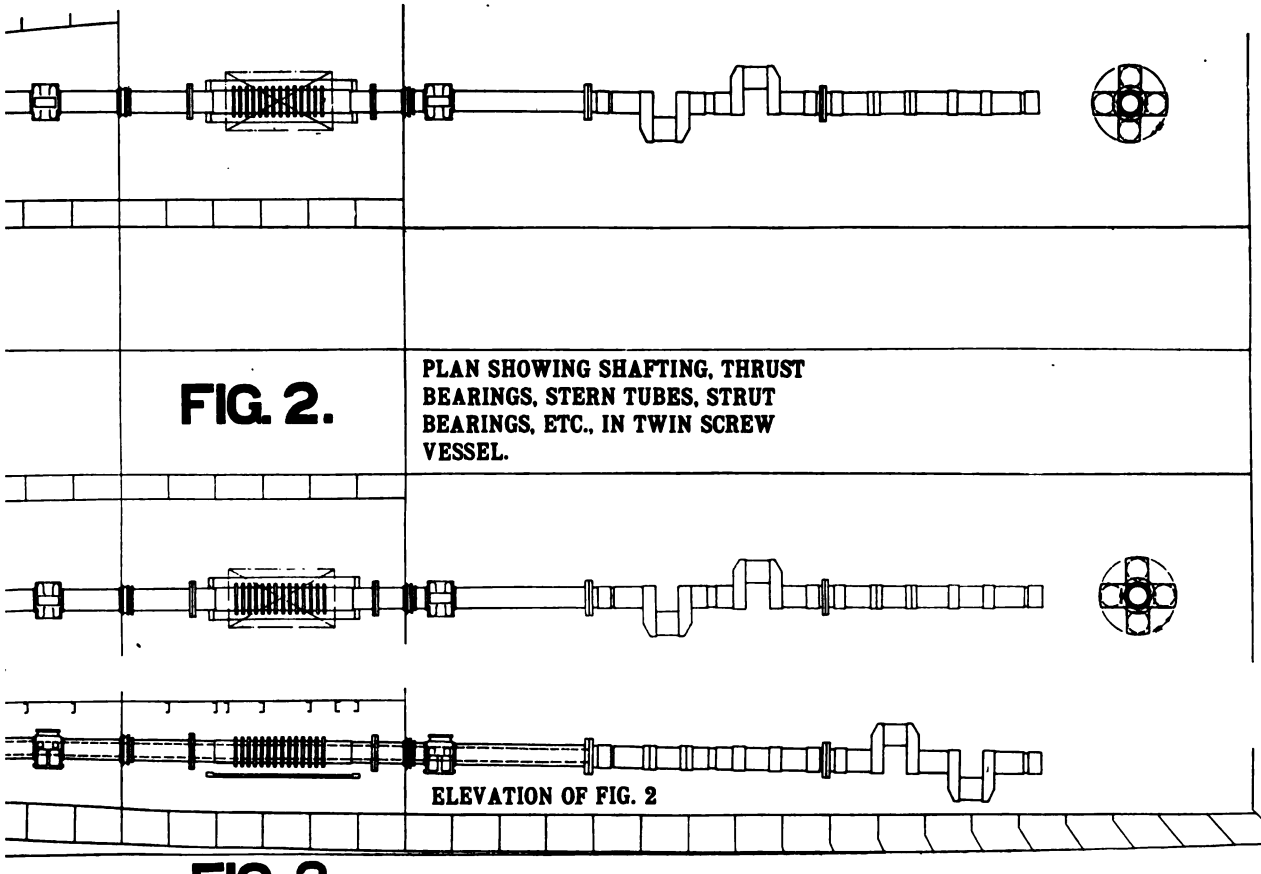


FIG. 2. PLAN SHOWING SHAFTING, THRUST BEARINGS, STERN TUBES, STRUT BEARINGS, ETC., IN TWIN SCREW VESSEL.

ELEVATION OF FIG. 2

FIG. 3.

PLATE 18.

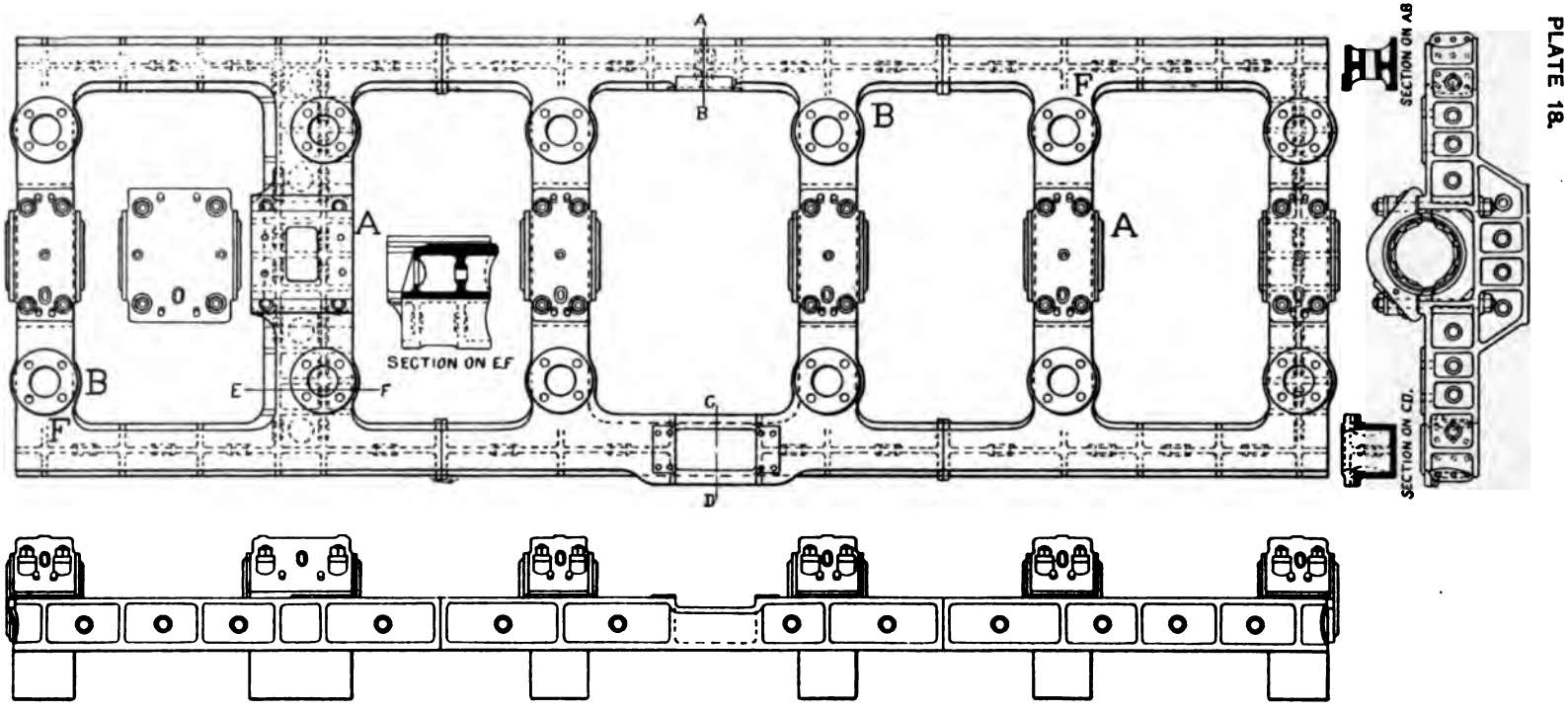
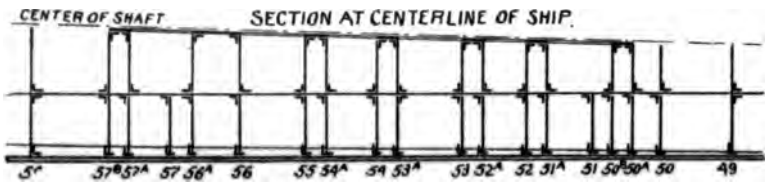


Fig. 1.



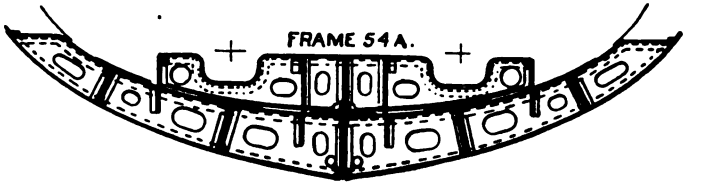


Fig. 2.

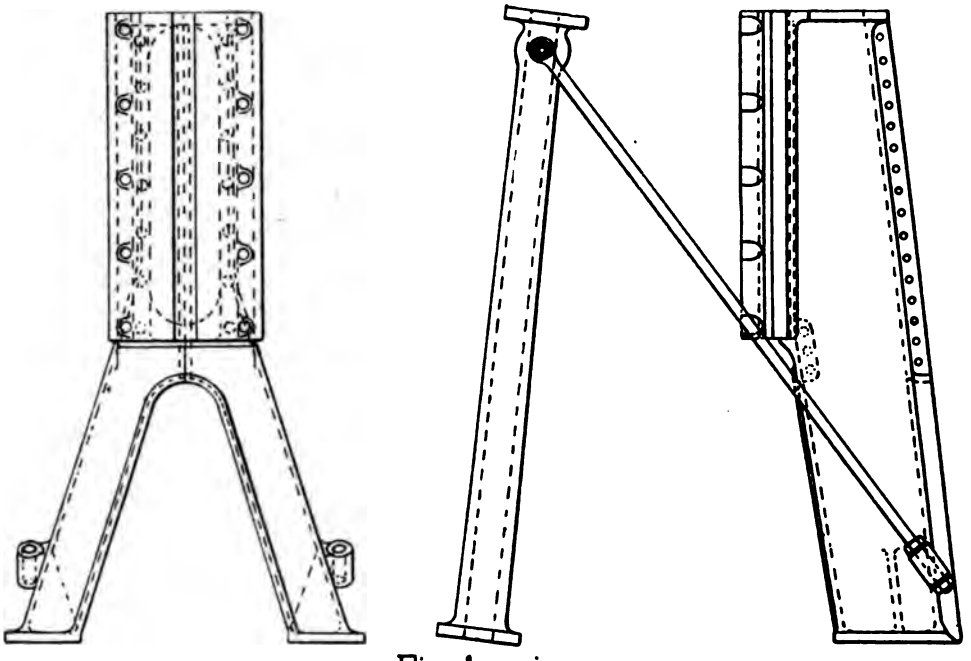
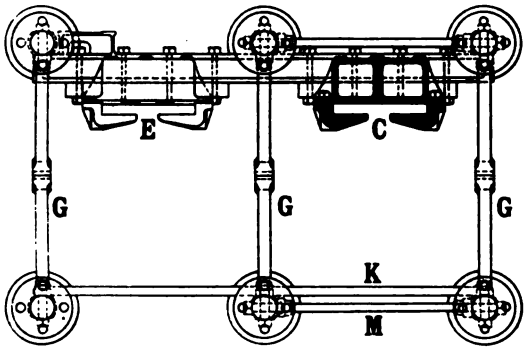
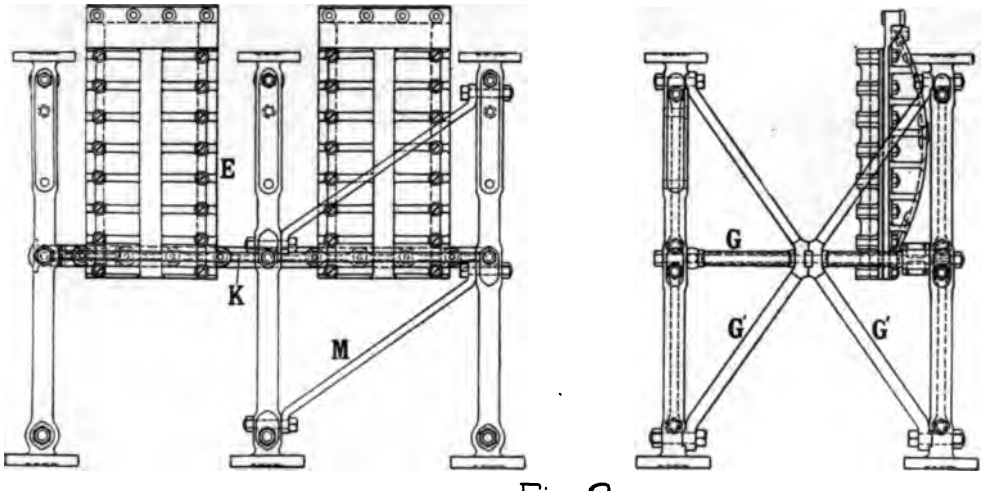


Fig. 1.





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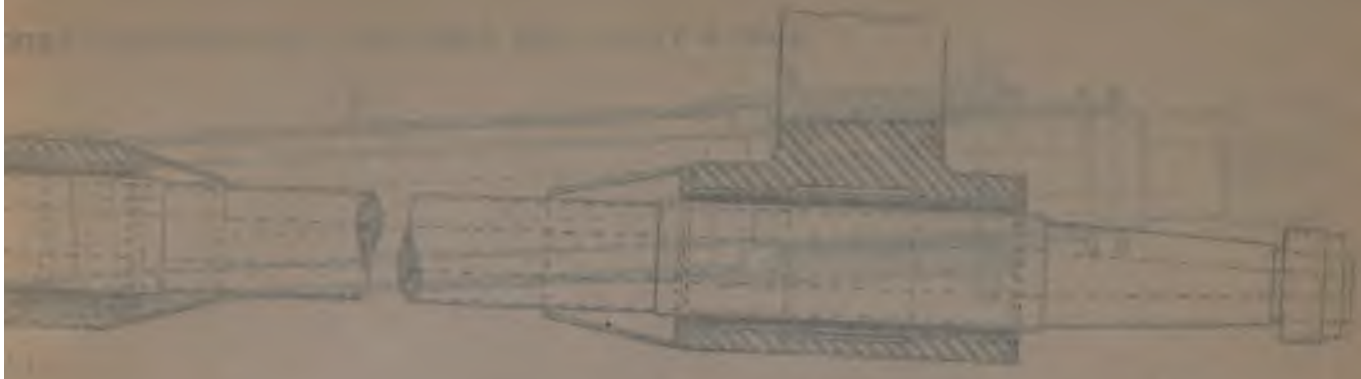


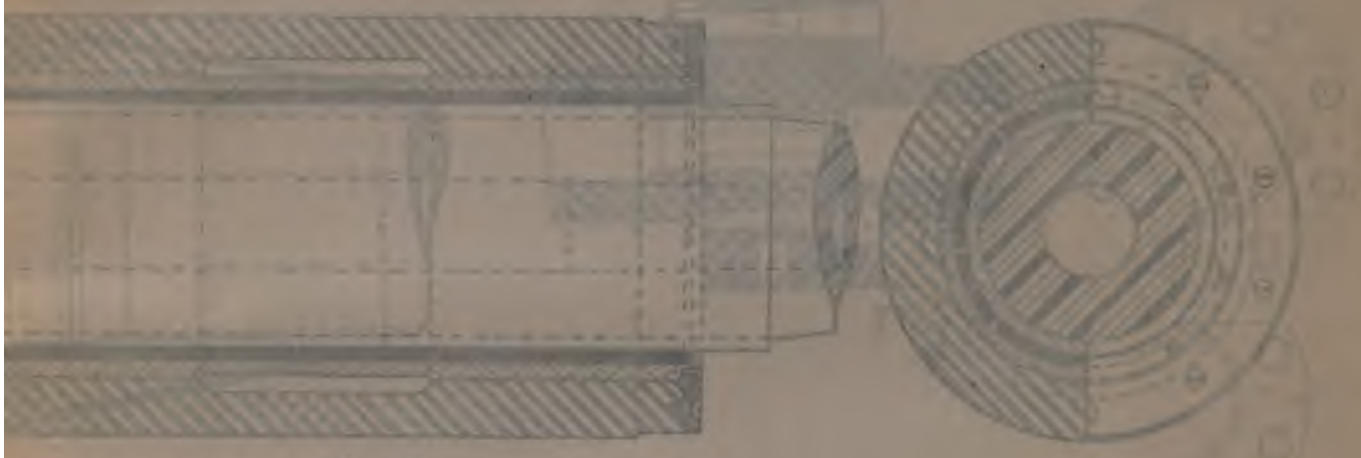


Fig 2

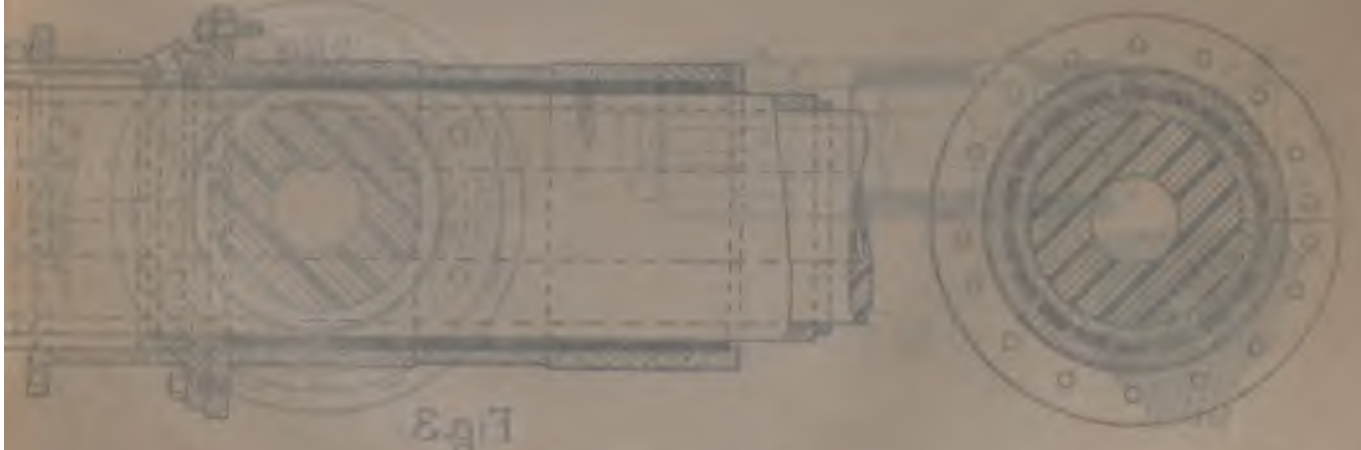


Fig 3

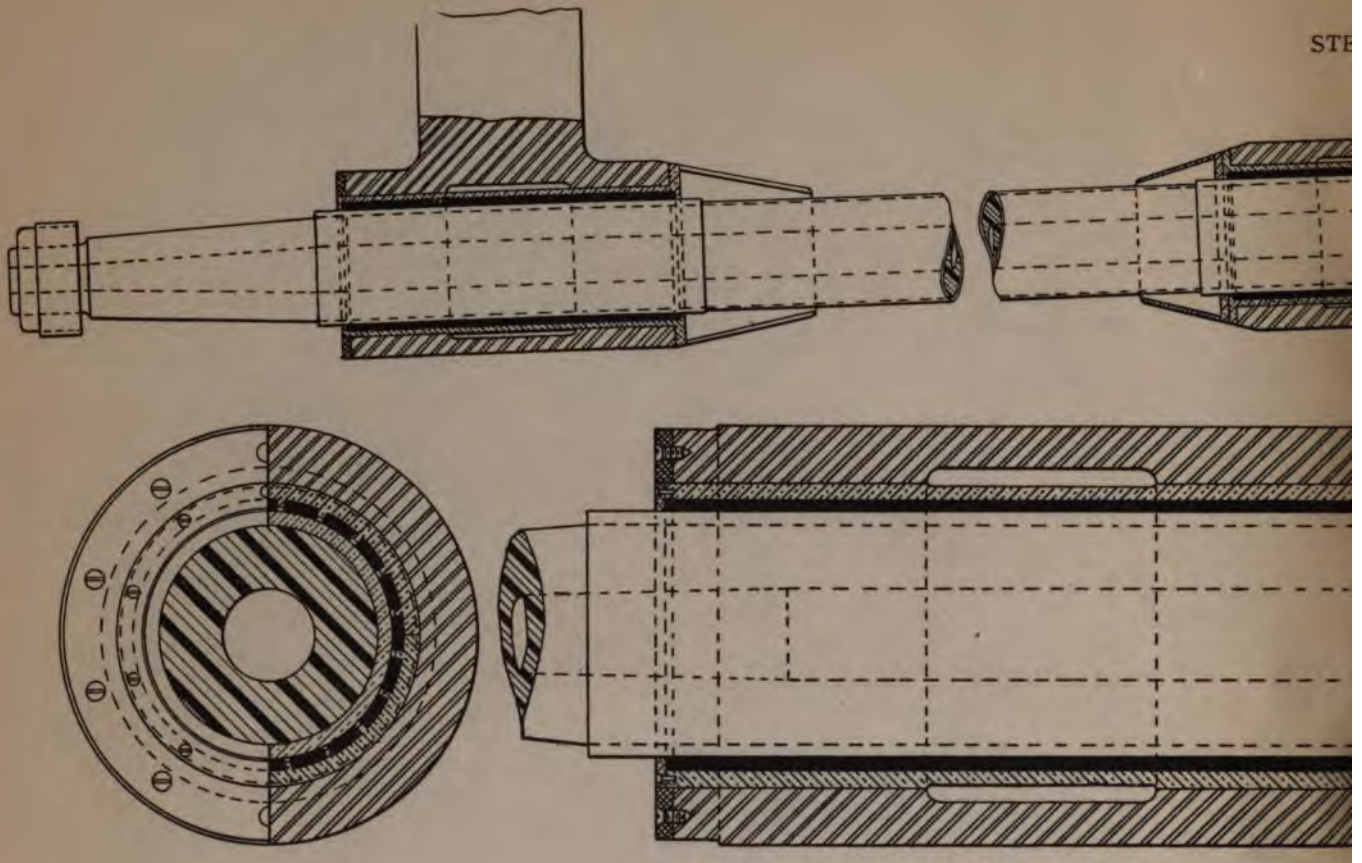


Fig 2.

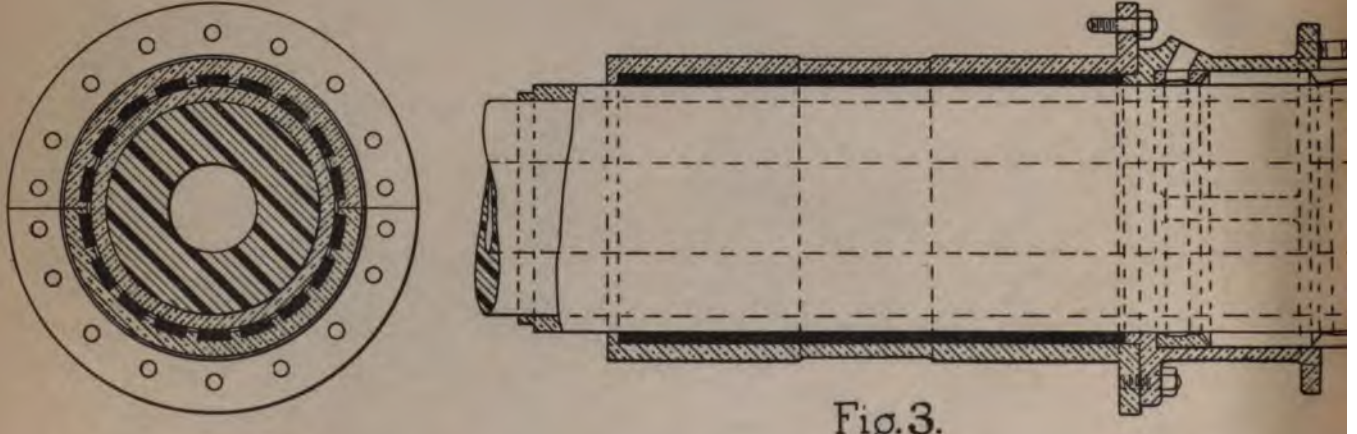
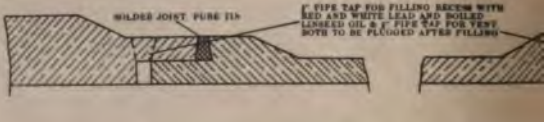
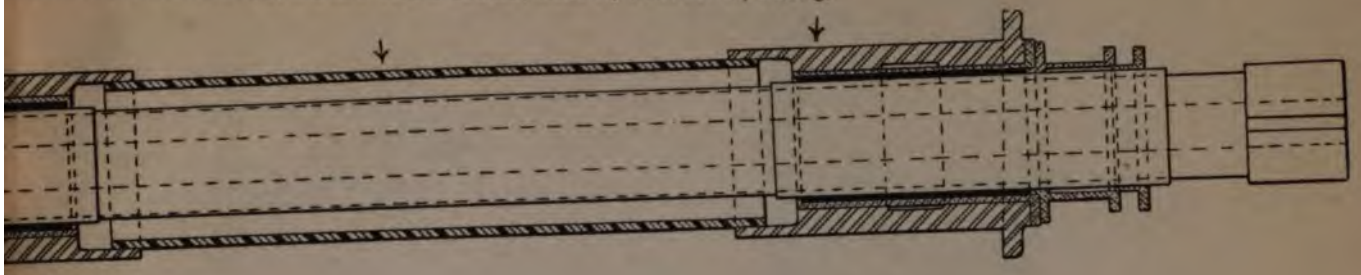


Fig. 3.


WILDER JOIST PURLIN
½" PIPE TAP FOR FILLING RECESS WITH RED AND WHITE LEAD AND BOILED LINSEED OIL & ½" PIPE TAP FOR VENT BOTH TO BE PLUGGED AFTER FILLING

TUBE AND BEARINGS, BUILT INTO HULL, SEE A, PLATE 16, FIG. 3.



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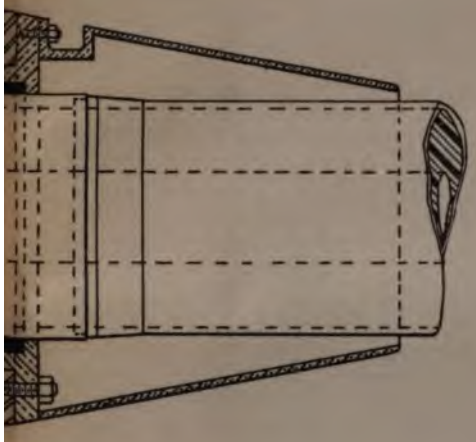


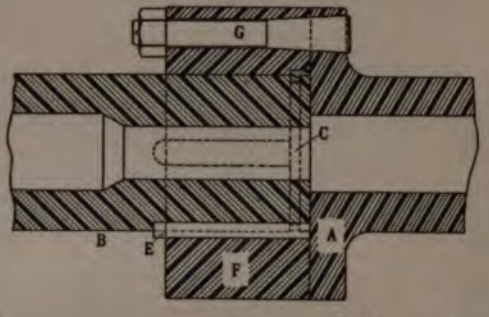
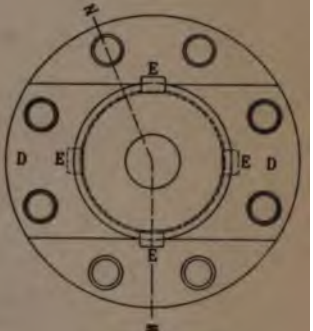
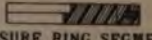
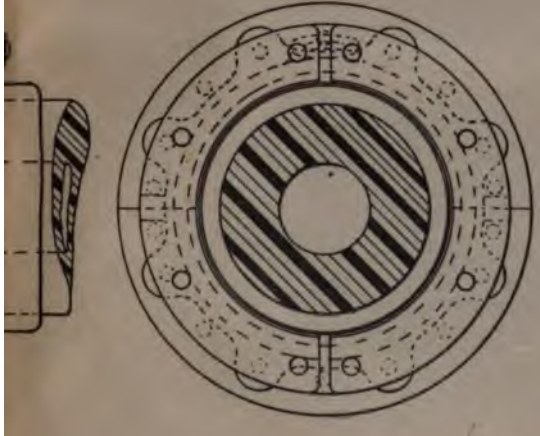


Fig.4.





PRESSURE RING SEGMENT



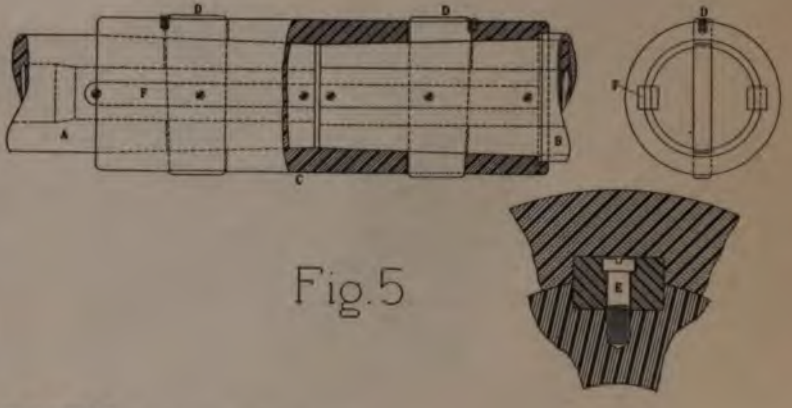


Fig.5

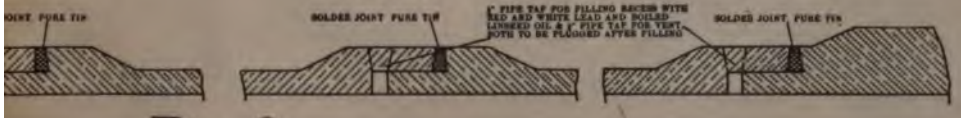


Fig.6.

UNIV. OF SYDNEY

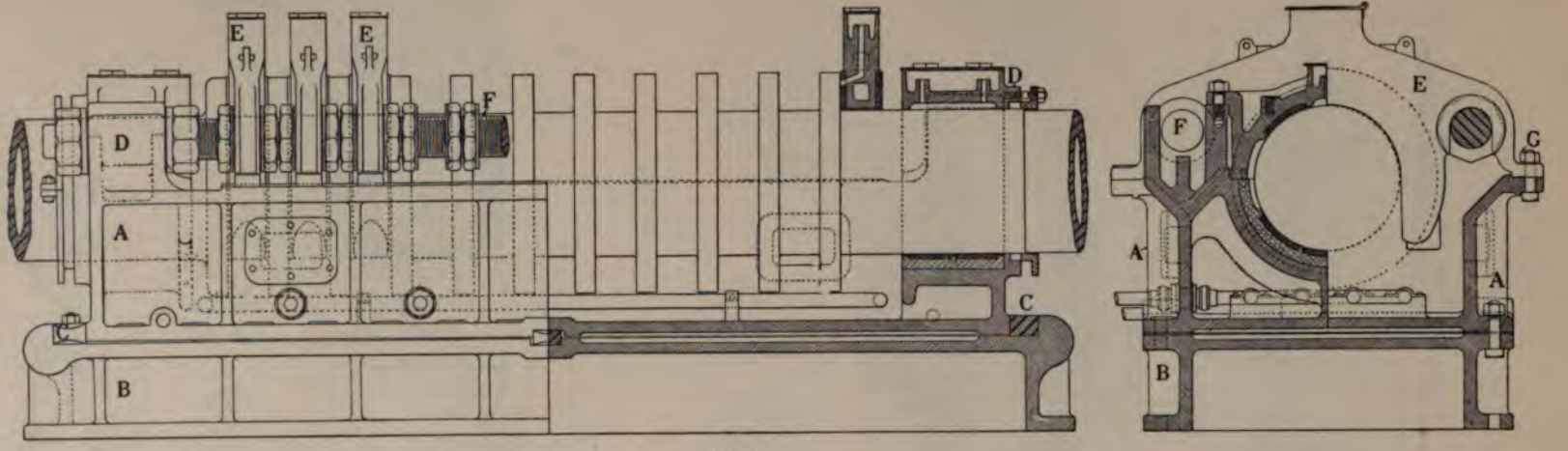


Fig.1

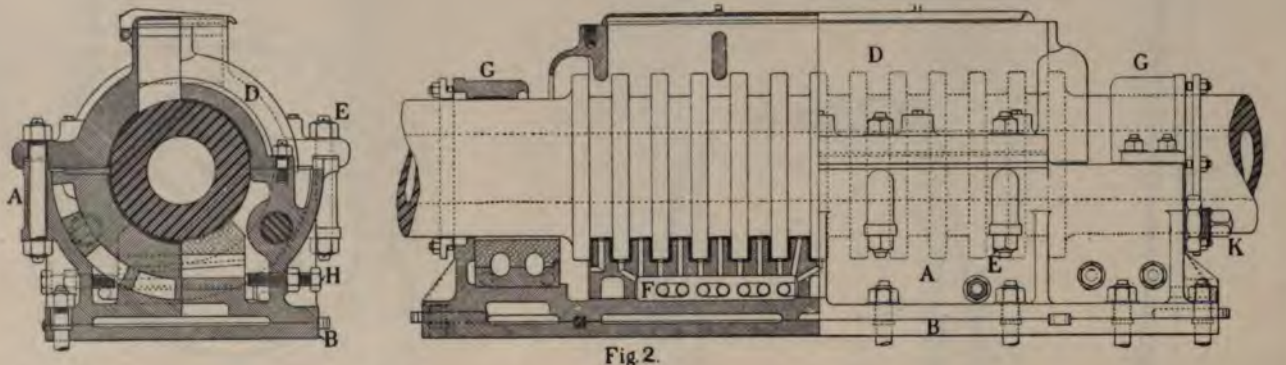
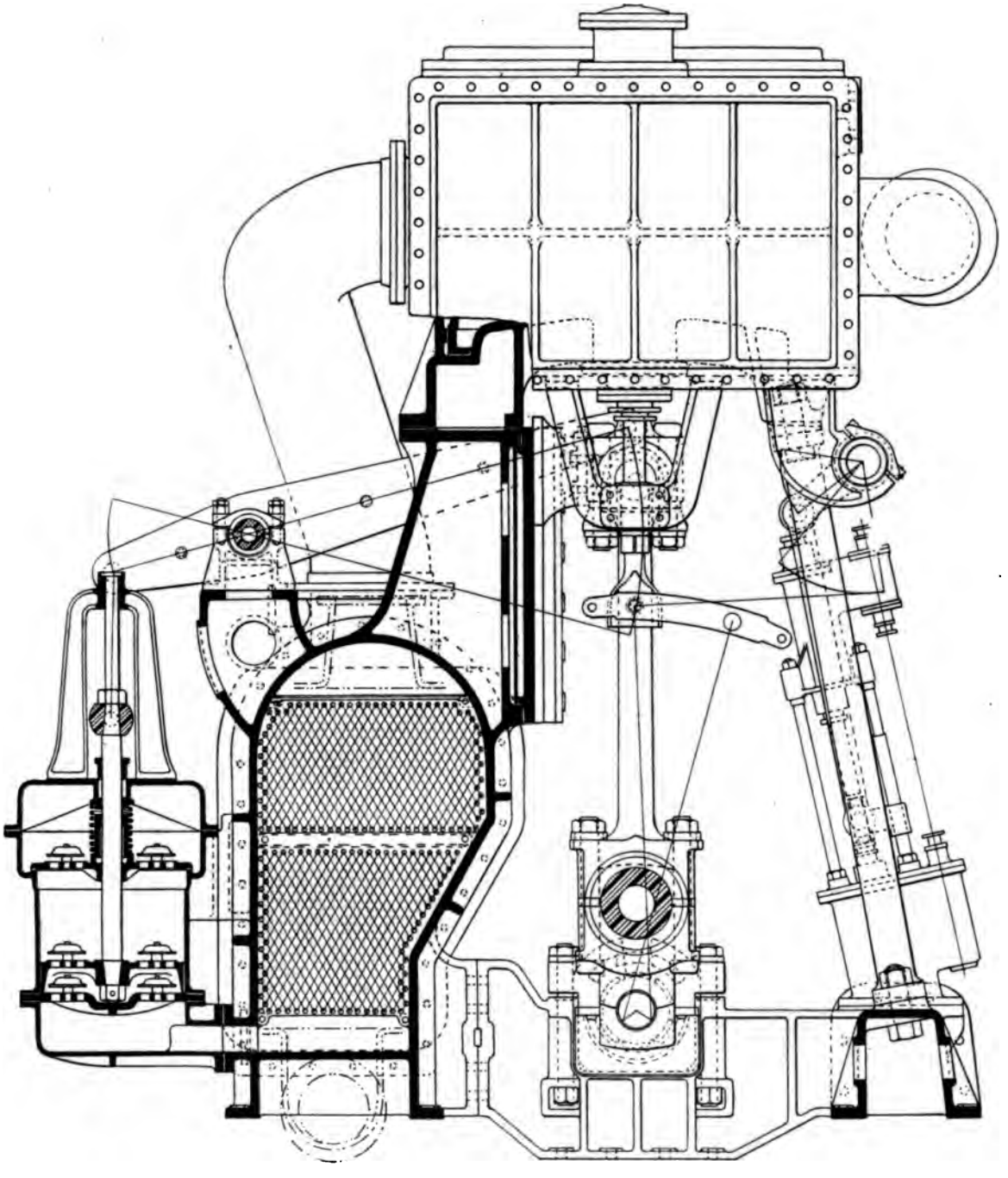
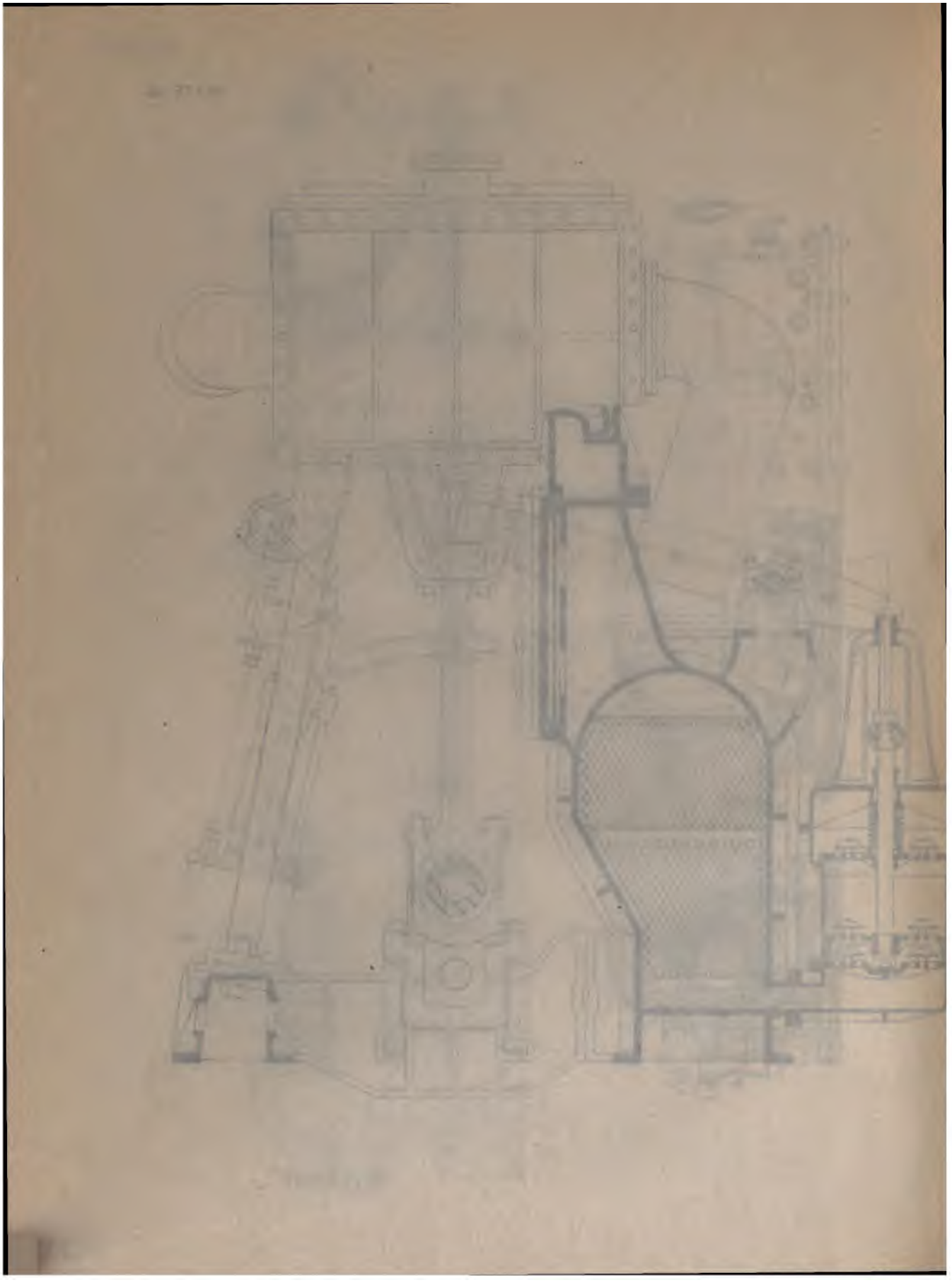
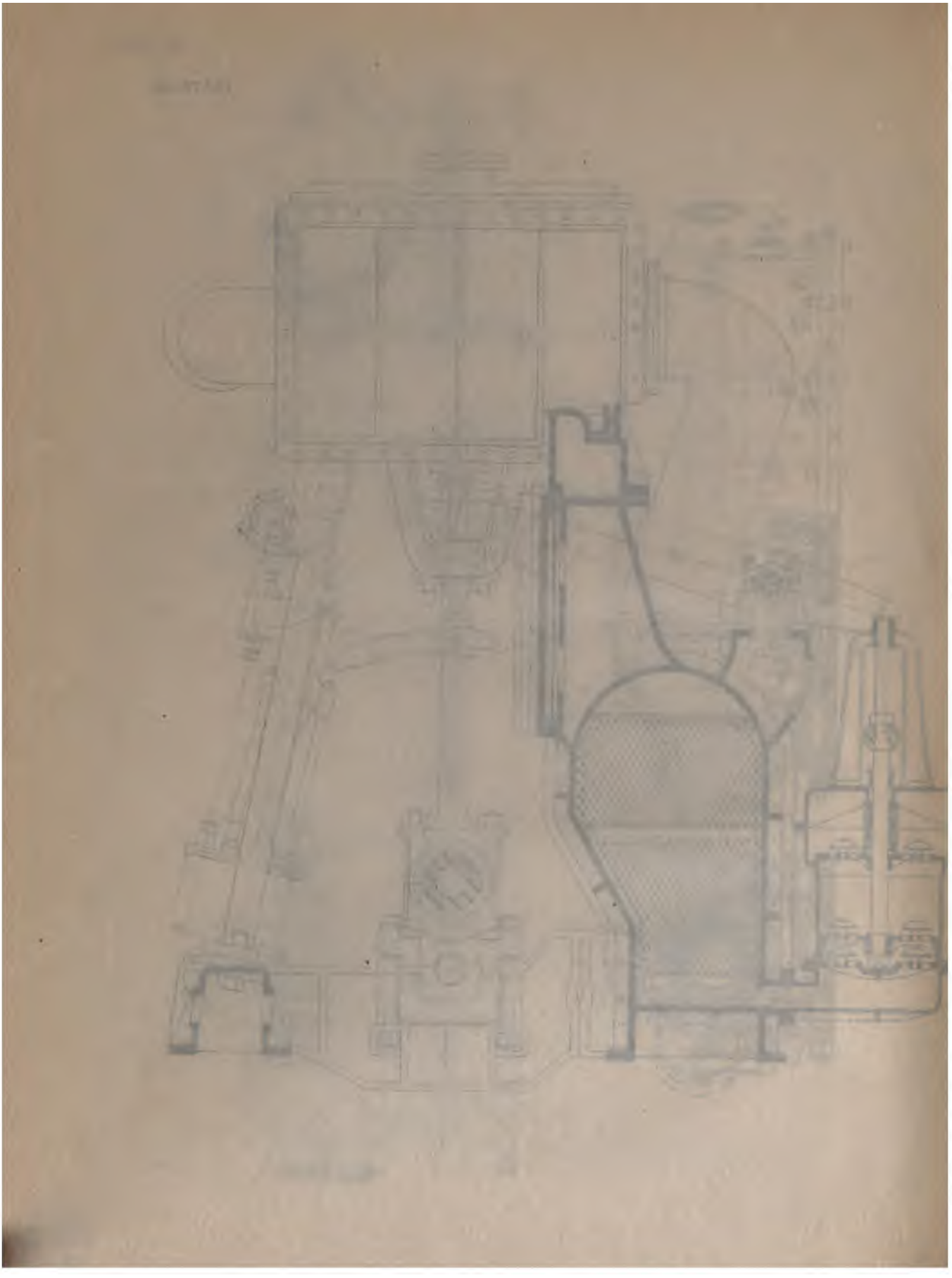


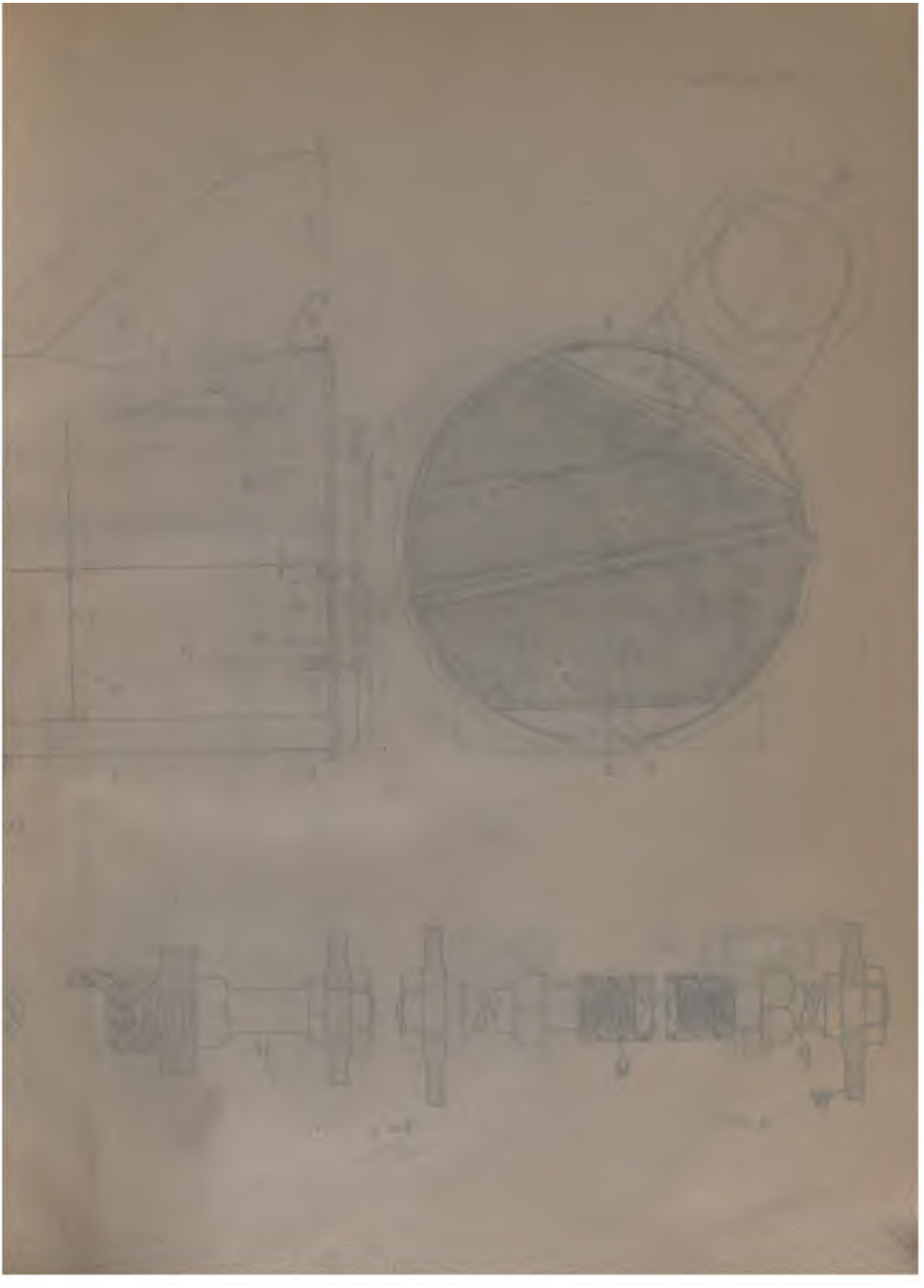
Fig.2.

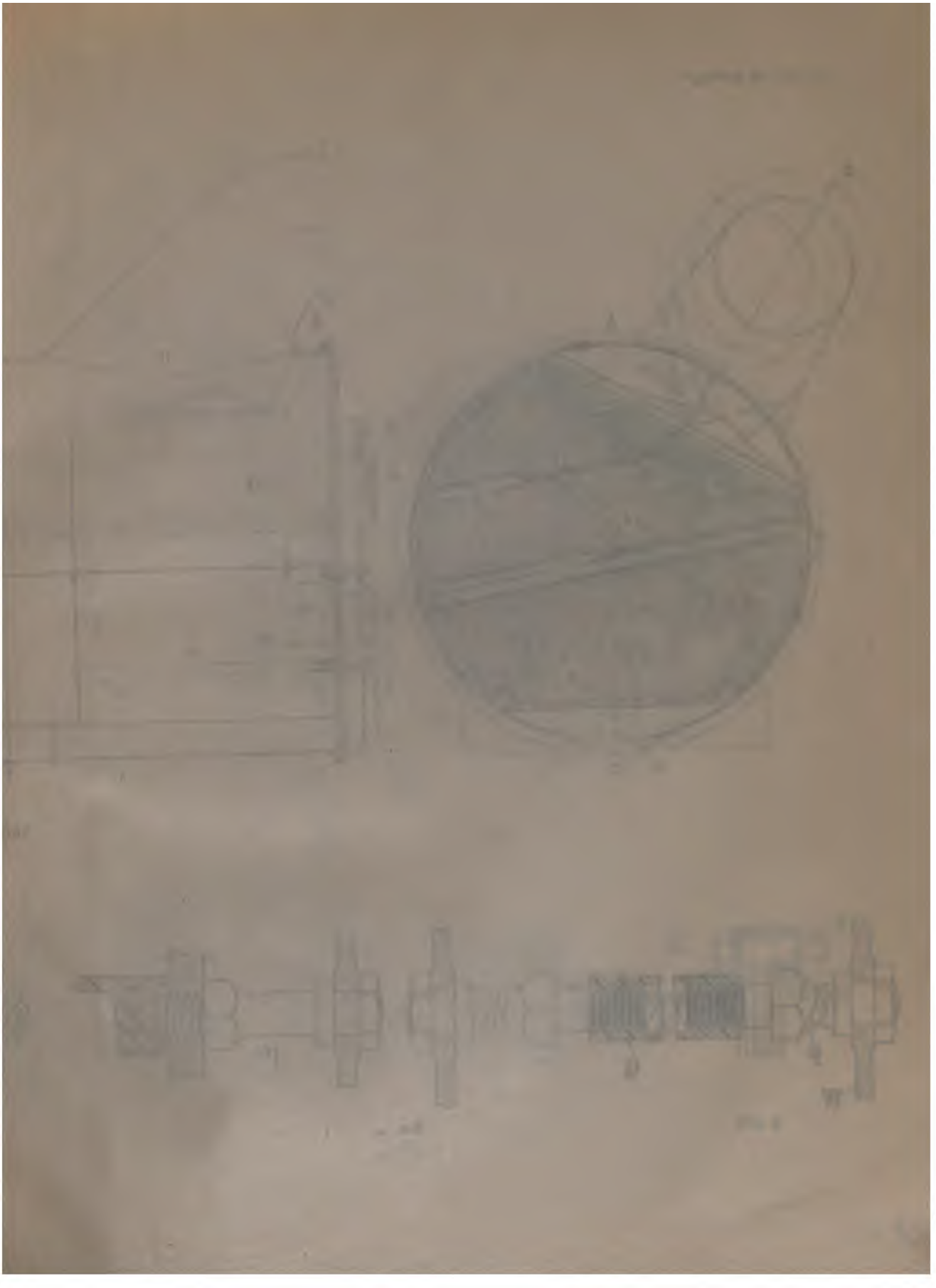
PLATE 23.

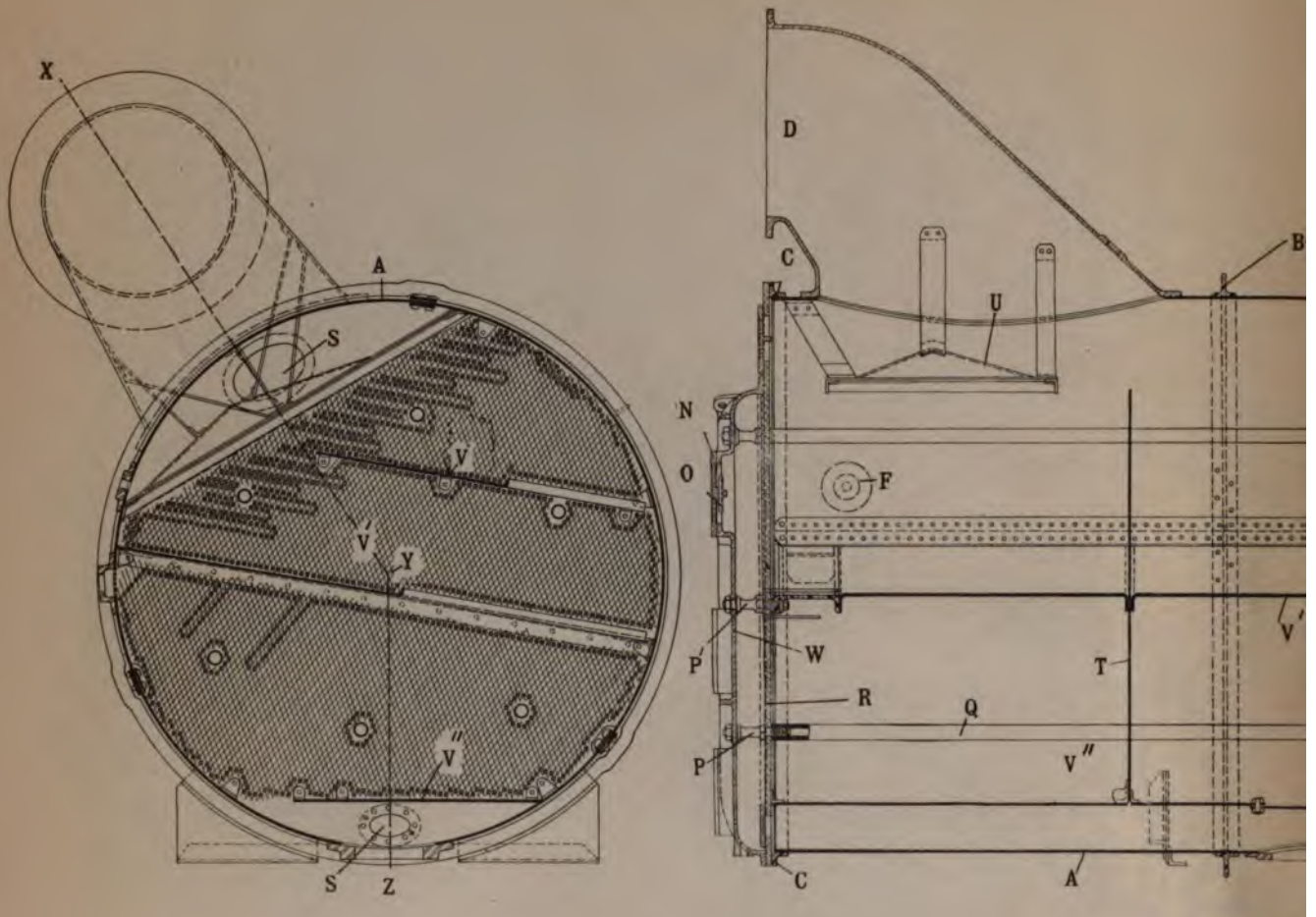












SECTION

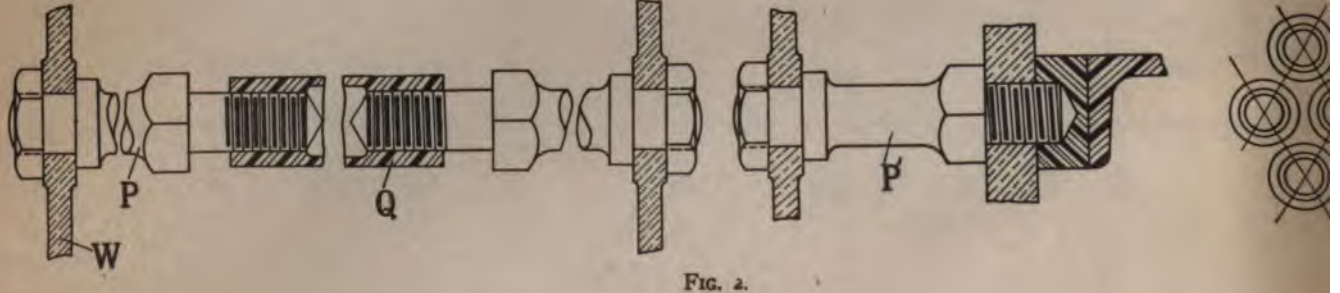
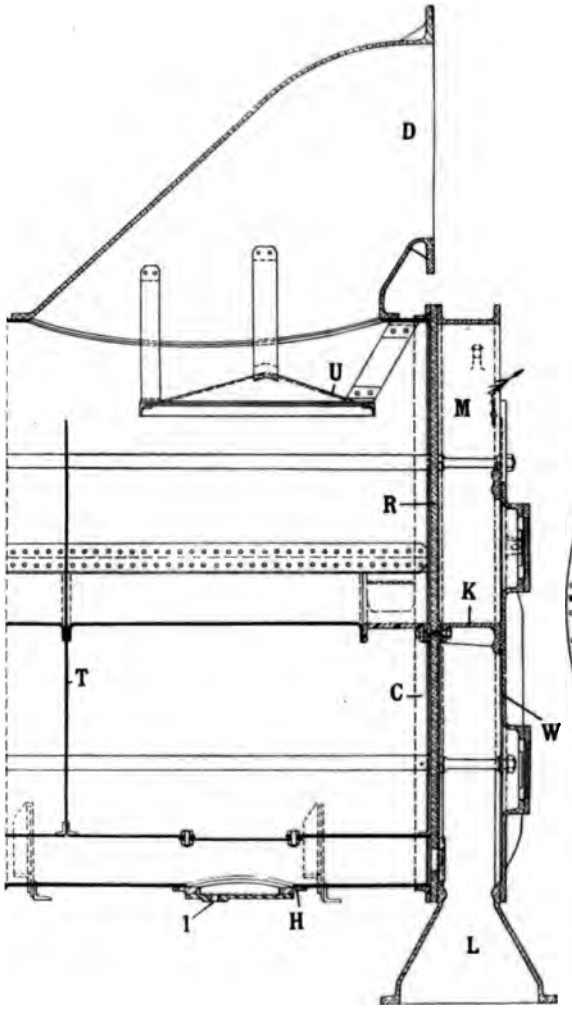
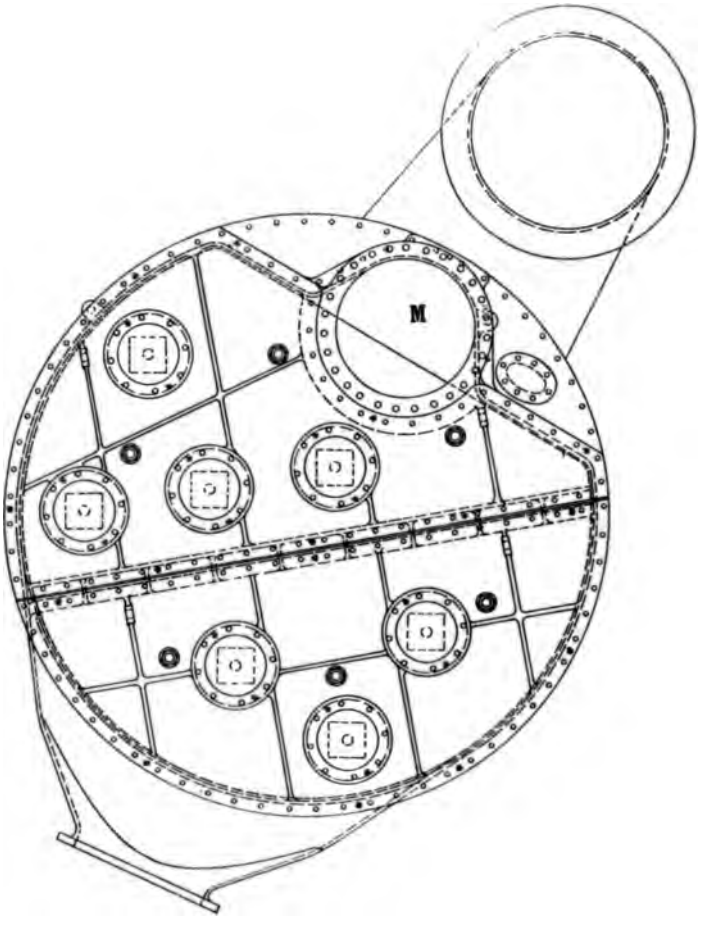


FIG. 2.





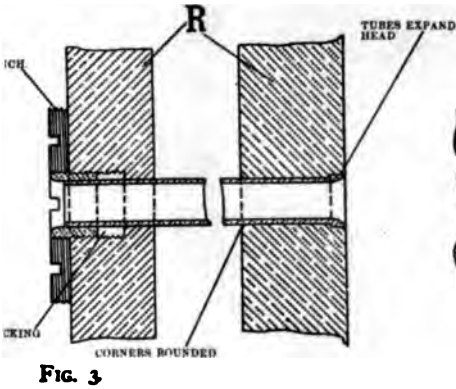


Fig. 3

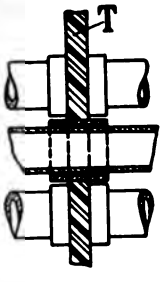
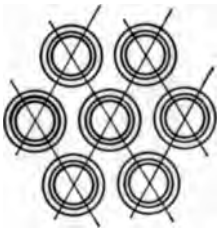


Fig. 4.



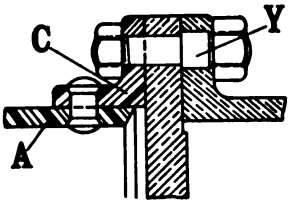


Fig. 5.

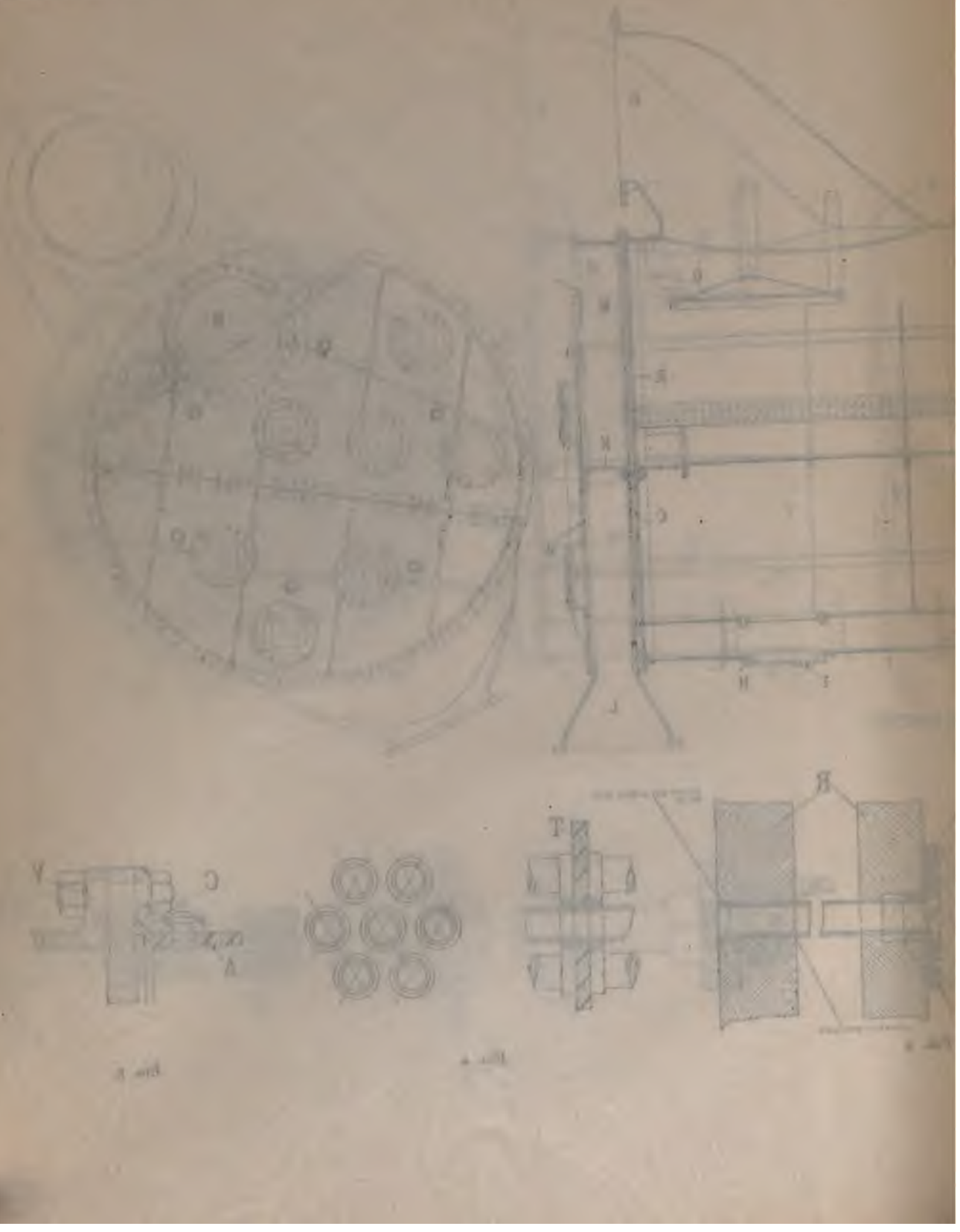


PLATE 26.

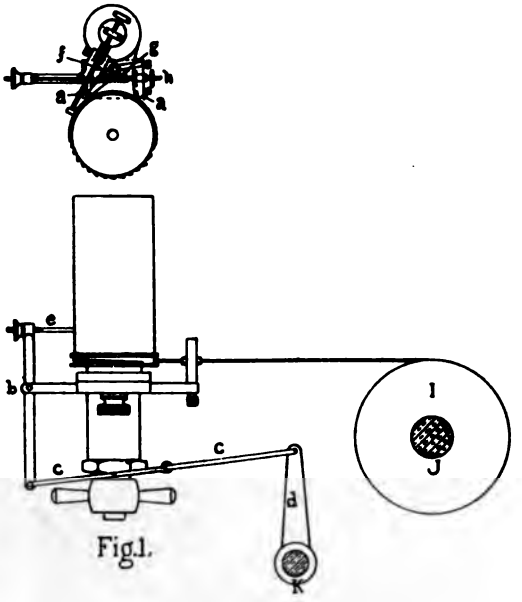


Fig.1.

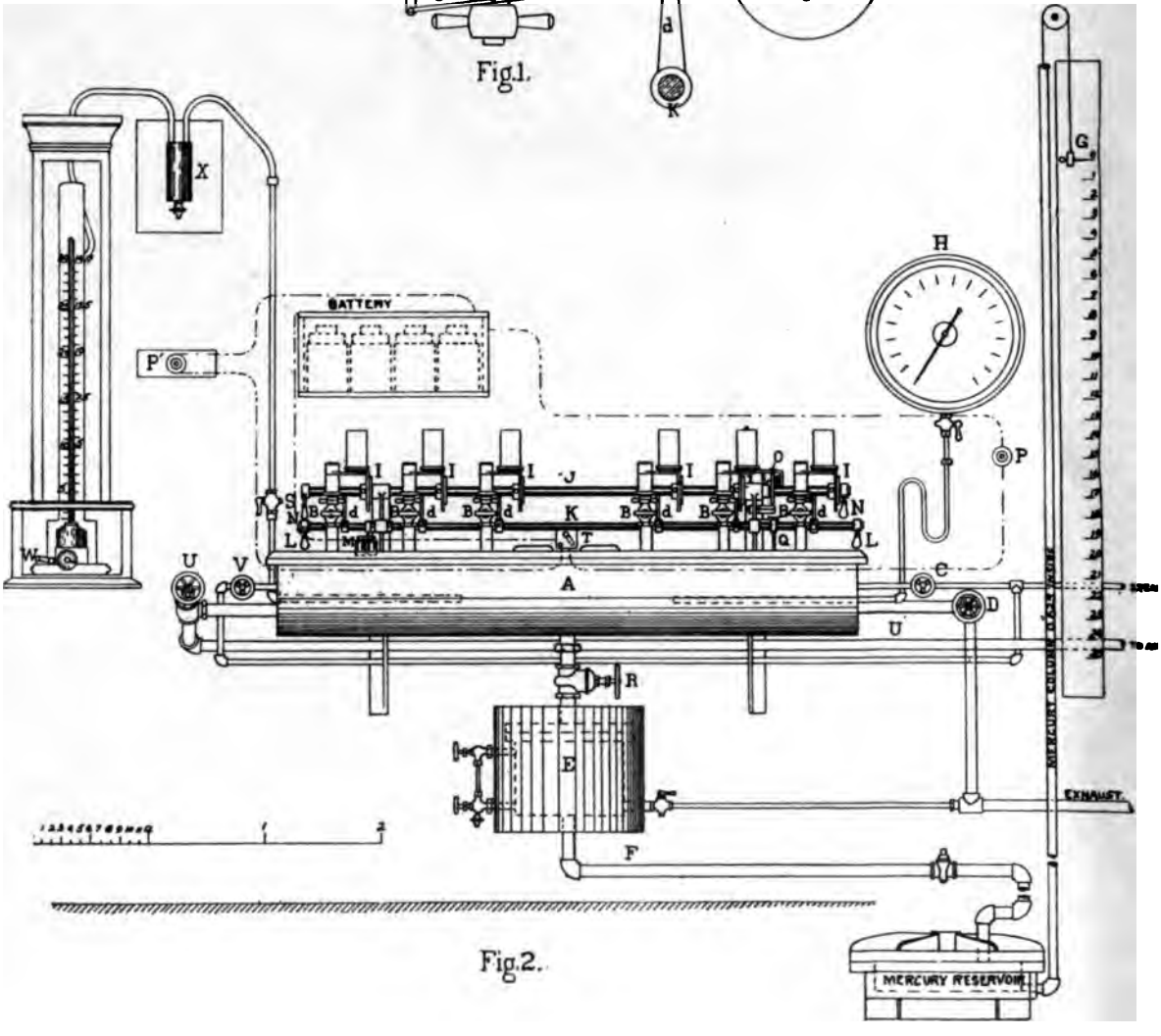


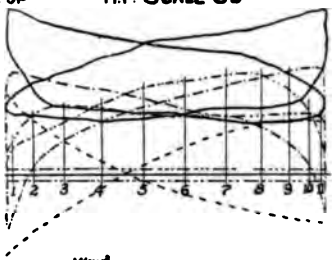
Fig.2.

CRANK EFFORT DIAGRAM

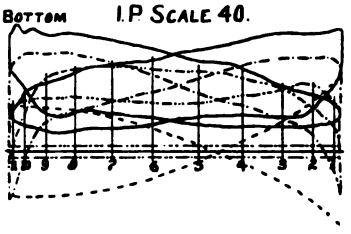
TO ACCOMPANY NOTES ON THE DESIGN OF PROPELLING MACHINERY FOR NAVAL VESSELS.
DEPARTMENT OF MARINE ENGINEERING, U.S. NAVAL ACADEMY.

—— ORIGINAL CARDS
-·-·- CORRECTED FOR BACK PRESSURE
----- INERTIA CURVES
_ _ _ ALL CORRECTIONS MADE FOR TOP END
—·—·— " " " " BOTTOM END

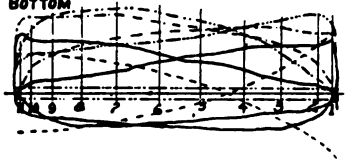
OP H.P. SCALE 80



BOTTOM I.P. SCALE 40.



L.P. SCALE 20
BOTTOM



$I = .00034 \frac{WrN^2}{A}$, PER SQ IN. OF PISTON

$I = .00034 \times \frac{6252 \times 8 \times 130^2}{804.25 \times 12} = 65.42$

AT TOP = $65.42 \times \frac{5}{4} = 81.75$

AT BOTTOM = $65.42 \times \frac{3}{4} = 49$

WT. PER SQ IN. = $\frac{5468}{804.25} = 6.5$

$= \frac{81.75}{2.2} = 37.16$

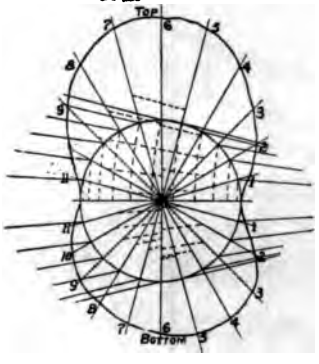
$= \frac{49}{2.2} = 22.27$

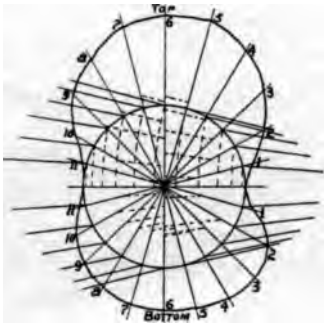
$= \frac{6.5}{2.2} = 2.93$

$= \frac{81.75}{52} = 15.72$

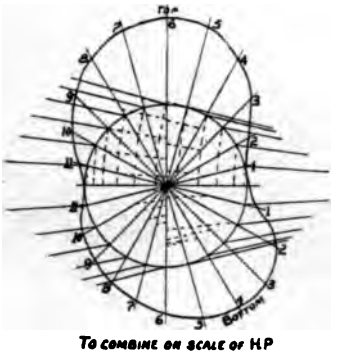
$= \frac{49}{52} = 9.42$

$= \frac{6.5}{52} = 1.25$

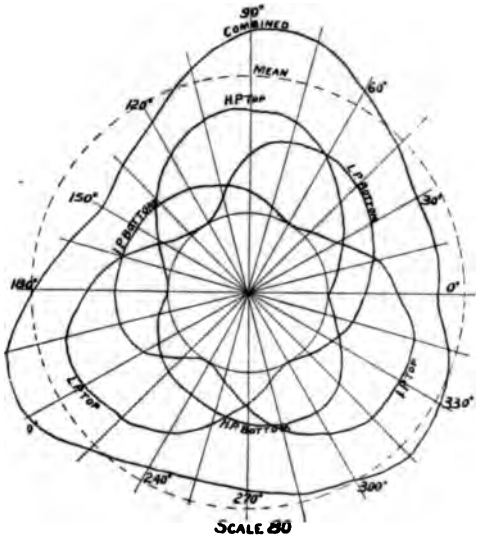


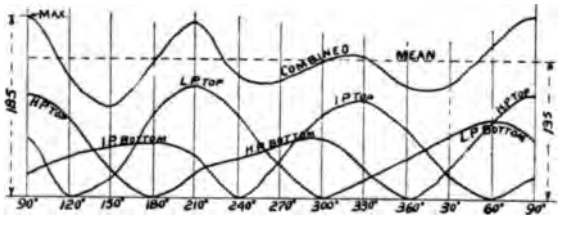


TO COMBINE ON SCALE OF H.P
MULTIPLY ALL ORDINATES
BY $2.2 \times \frac{40}{80} = 1.1$



TO COMBINE ON SCALE OF H.P
MULTIPLY ALL ORDINATES
BY $5.2 \times \frac{20}{80} = 1.3$





MAX T.F. = 804.25 × 185 = 148786.25 POUNDS
MEAN T.F. = 804.25 × 135 = 108573.75 "
RATIO OF MAX TO MEAN = $\frac{185}{135} = 1.37$

U.S.S. BROOKLYN.

DATA:

AREA OF H.P. PISTON 804.25 SQ INS
RATIO OF PISTON AREAS 1 : 2.2 : 5.2
WEIGHT OF RECIPROCATING PARTS 5252 LBS.
STROKE 42" CONNECTING ROD 84"
REVOLUTIONS 130 PER MINUTE
VERTICAL. TRIPLE EXPANSION ENGINE
CRANKS AT 120°

PLATE 28.

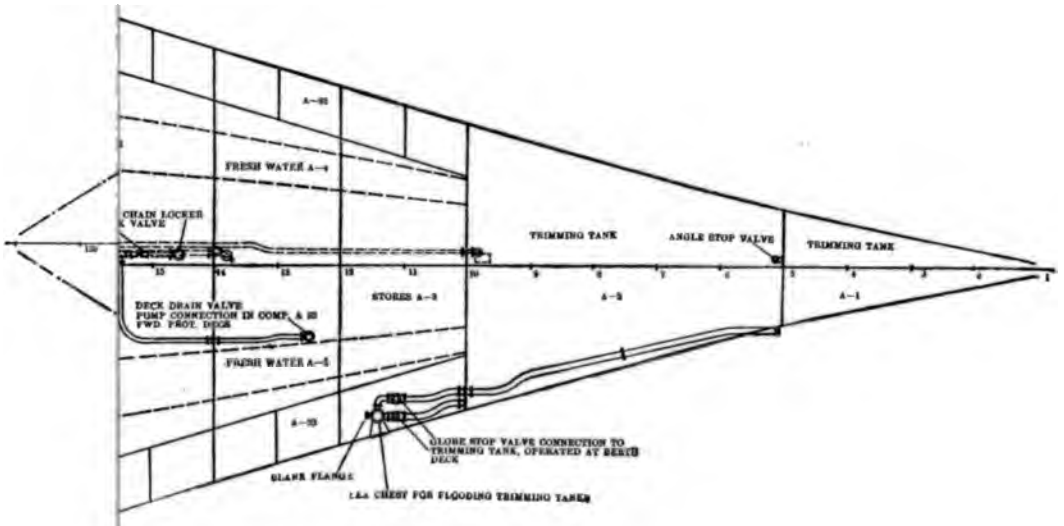

FRESH WATER A—4
CHAIN LOCKER & VALVE
TRIMMING TANK
ANGLE STOP VALVE
TRIMMING TANK
STORES A—2
A—2
A—1
DECK DRAIN VALVE
PUMP CONNECTION IN COMP. A 22
FWD. FROT. DECK
FRESH WATER A—5
A—23
GLOBE STOP VALVE CONNECTION TO TRIMMING TANK, OPERATED AT BRIDGE DECK
BLANK FLANGE
CASTINGS FOR FLOODING TRIMMING TANKS

PLATE 28.

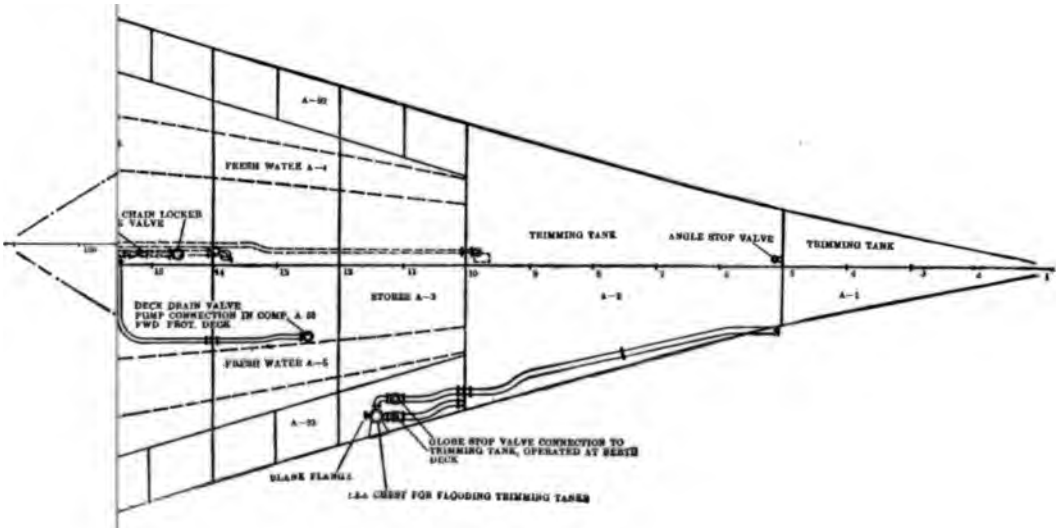
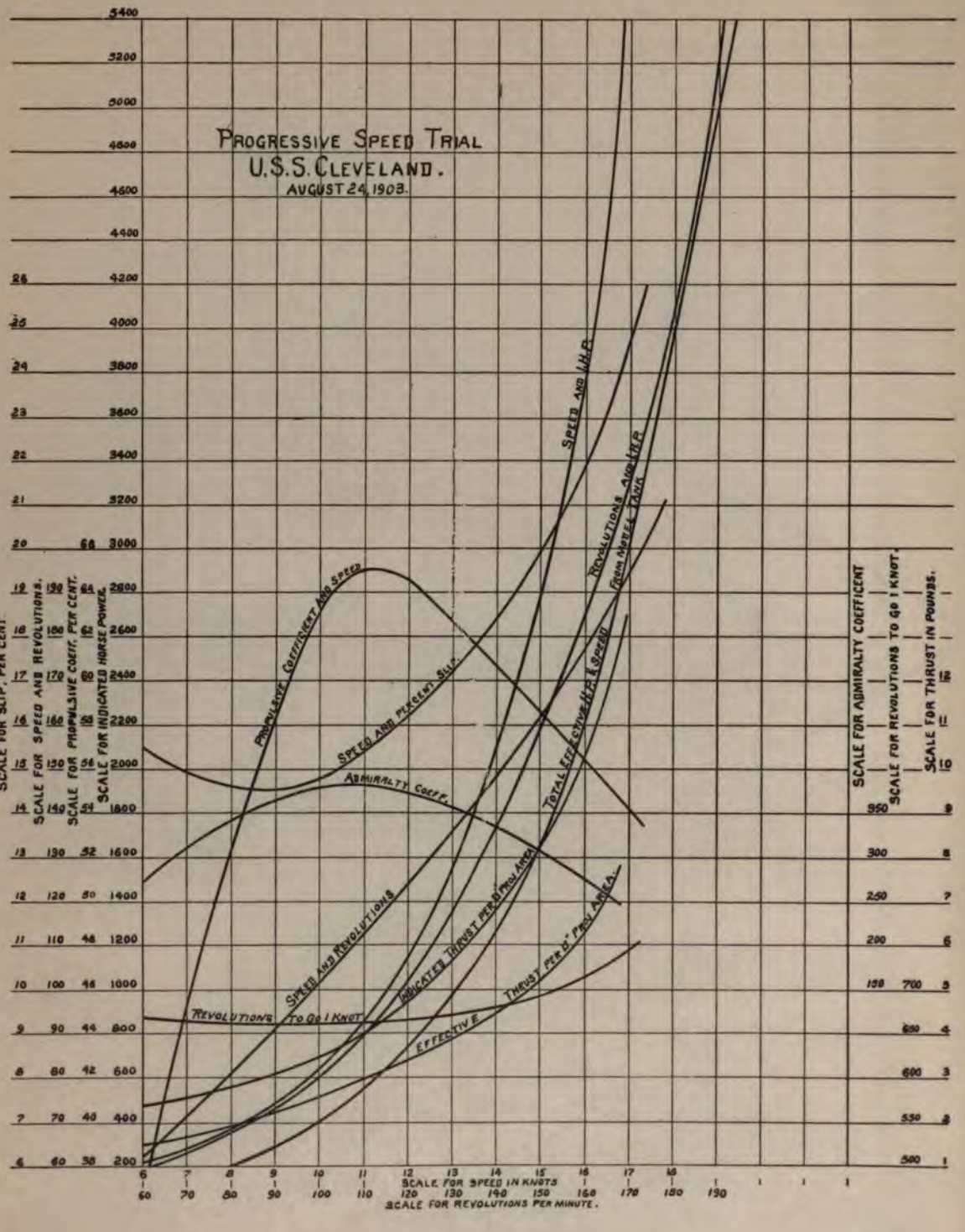
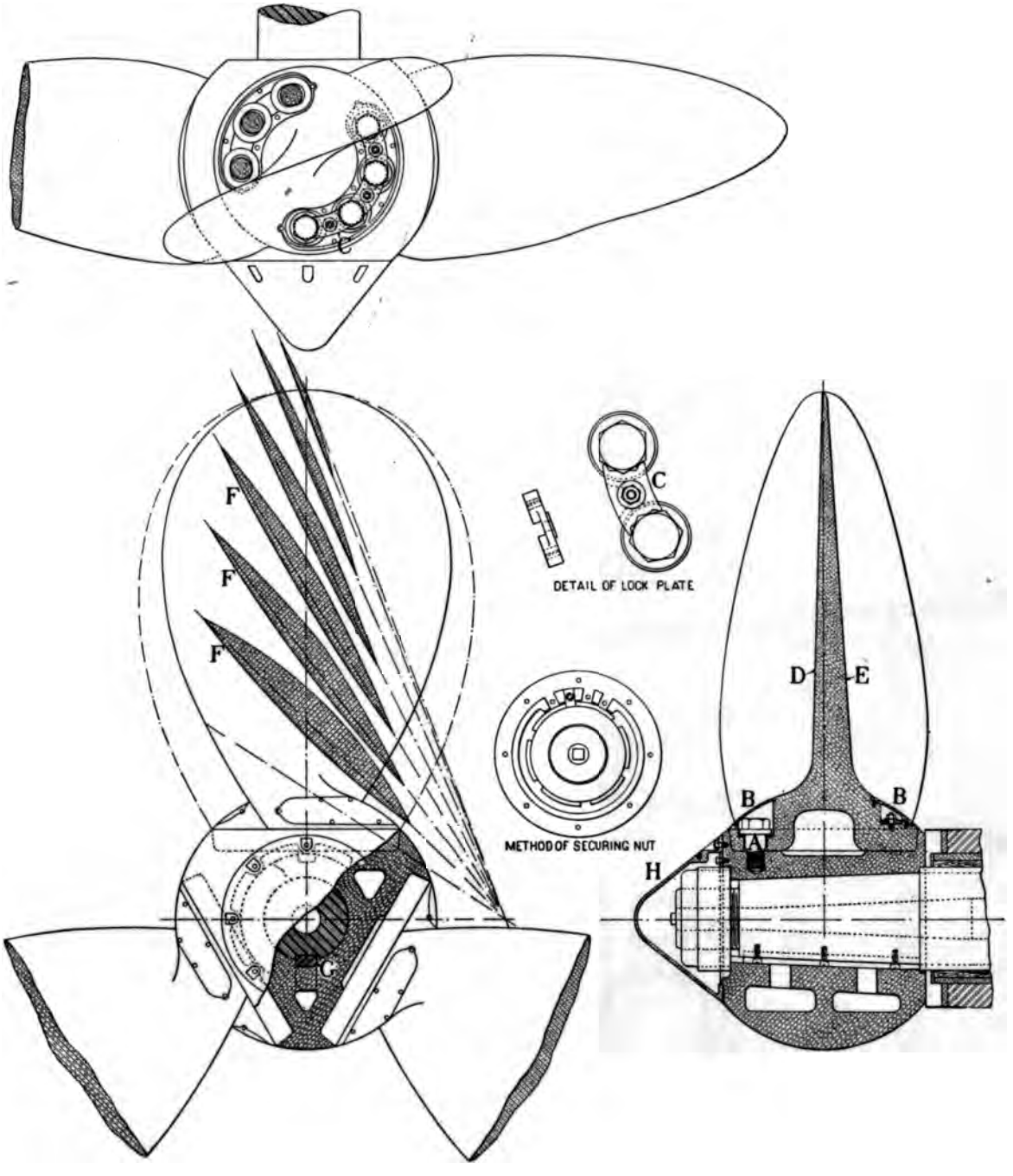
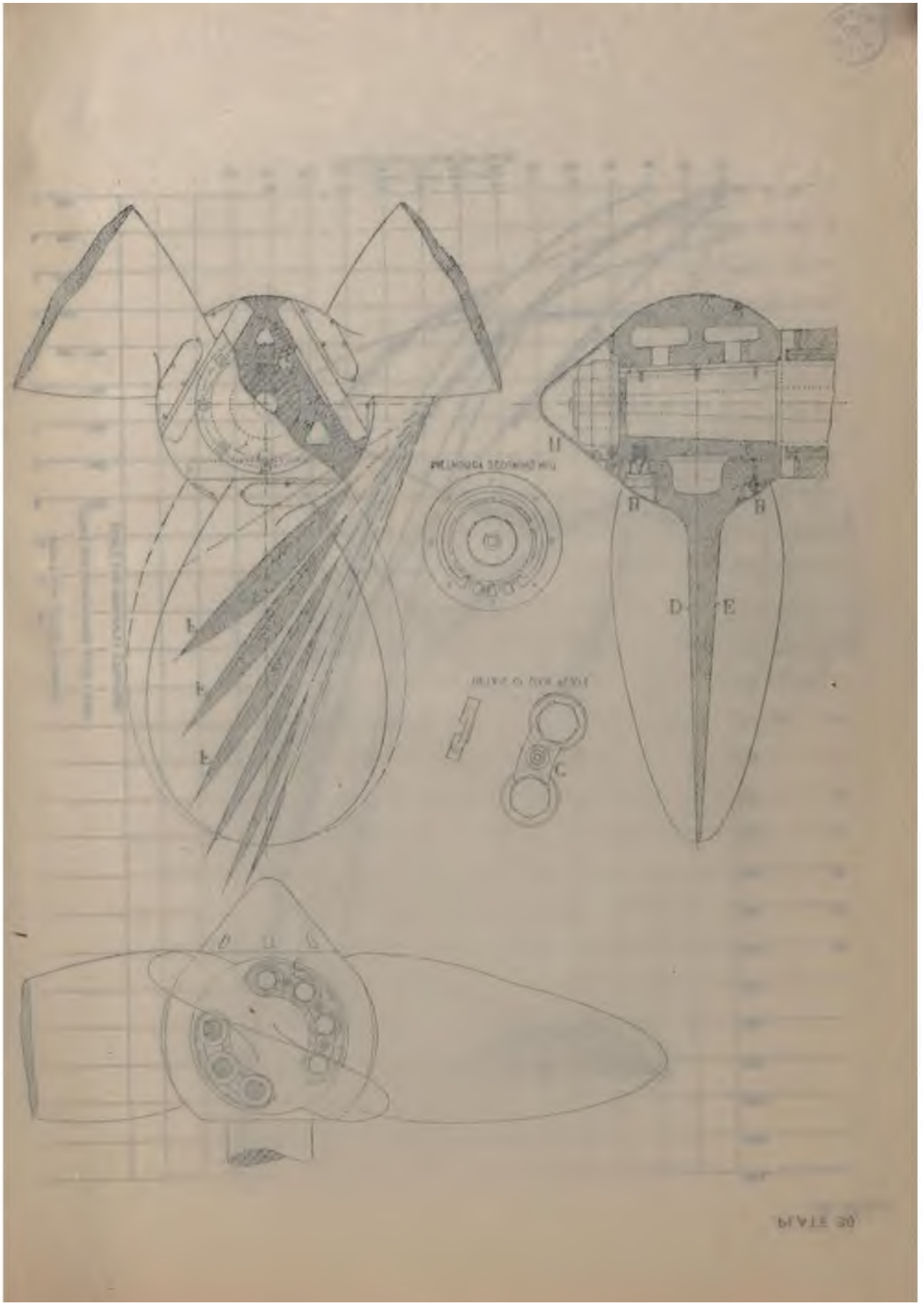
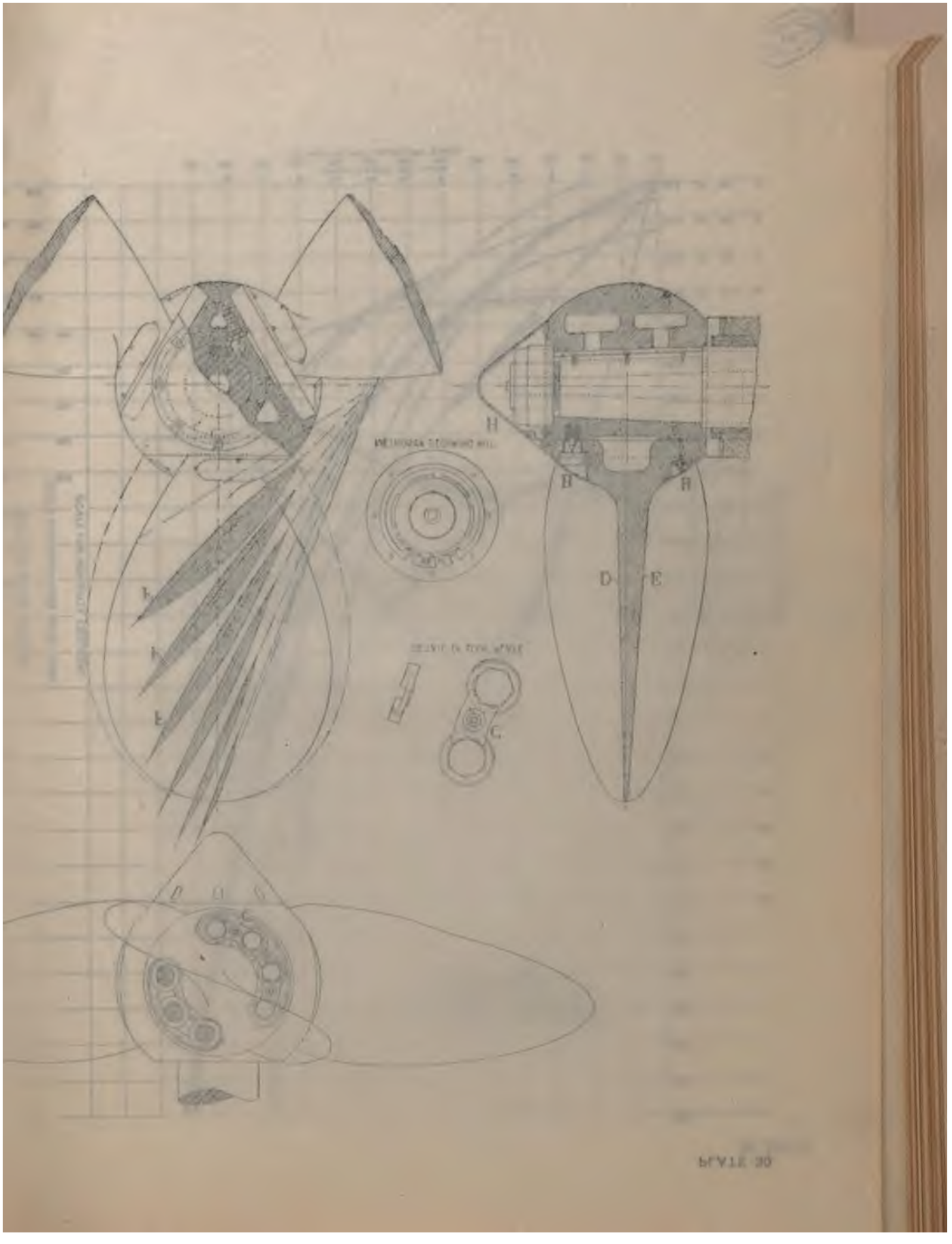

A—22
FRESH WATER A—4
CHAIN LOCKER
VALVE
TRIMMING TANK
ANGLE STOP VALVE
TRIMMING TANK
STORES A—3
A—2
A—1
DECK DRAIN VALVE
PUMP CONNECTION IN COMP. A 22
FWD. FROT. DECK.
FRESH WATER A—5
A—23
GLOBE STOP VALVE CONNECTION TO
TRIMMING TANK, OPERATED AT
DECK
BLANK FLANGE
CHEST FOR FLOODING TRIMMING TANKS

PLATE 29.

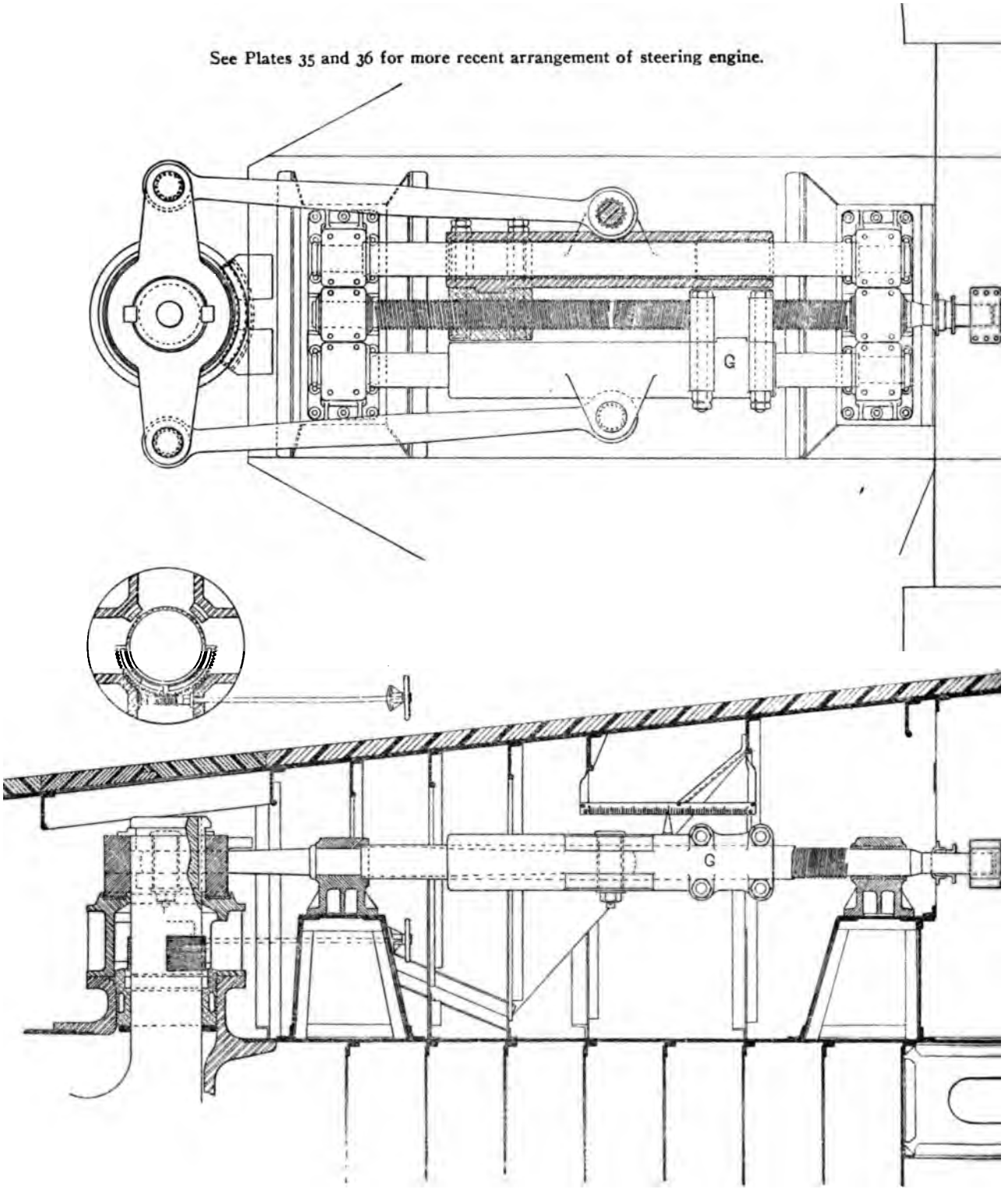

PROGRESSIVE SPEED TRIAL
U.S.S. CLEVELAND.
AUGUST 24, 1903.
SCALE FOR SLIP, PER CENT.
SCALE FOR SPEED AND REVOLUTIONS.
SCALE FOR PROPULSIVE COEFF. PER CENT.
SCALE FOR INDICATED HORSE POWER.
SCALE FOR ADMIRALTY COEFFICENT
SCALE FOR REVOLUTIONS TO GO 1 KNOT.
SCALE FOR THRUST IN POUNDS.
PROPULSIVE COEFFICIENT AND SPEED
SPEED AND PER CENT. SLIP.
ADMIRALTY COEFF.
SPEED AND I.H.P.
REVOLUTIONS AND I.H.P. FROM MODEL TANK
TOTAL EFFECTIVE H.P. & SPEED
SPEED AND REVOLUTIONS
REVOLUTIONS TO GO 1 KNOT
INDICATED THRUST PER □" PROJ. AREA
THRUST PER □" PROJ. AREA.
EFFECTIVE
SCALE FOR SPEED IN KNOTS
SCALE FOR REVOLUTIONS PER MINUTE.

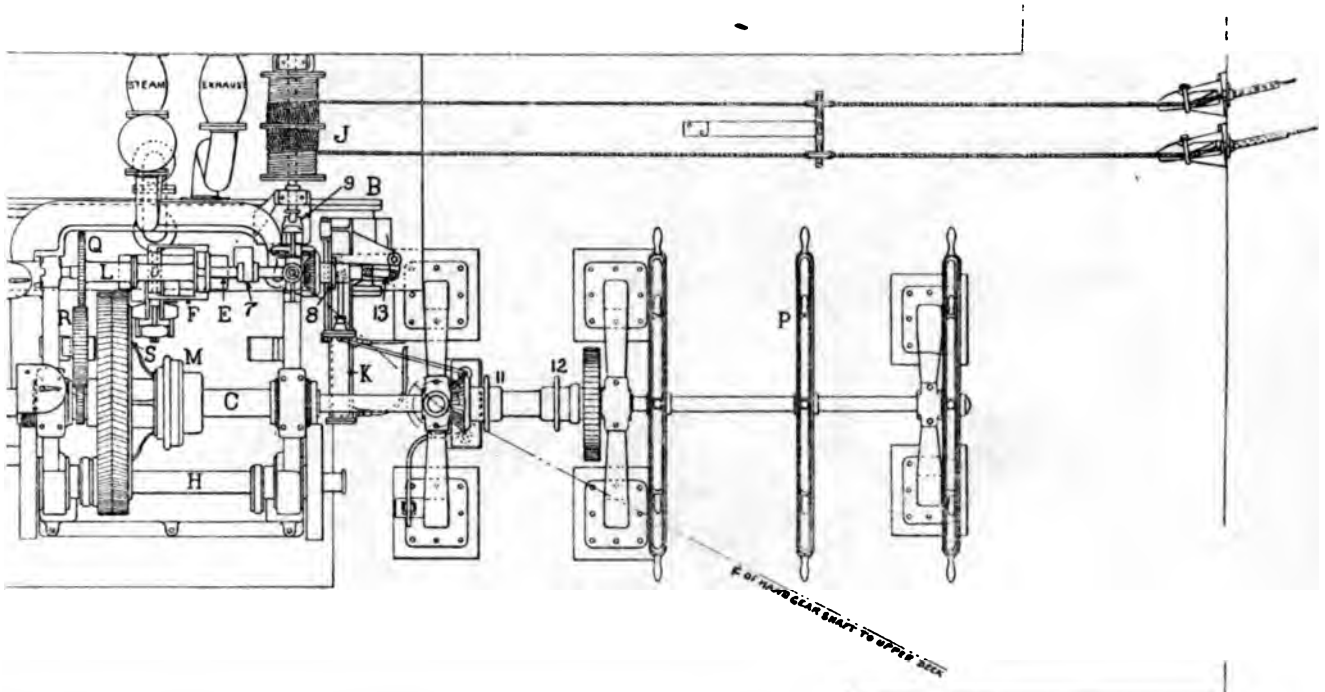

F
F
F
C
DETAIL OF LOCK PLATE
METHOD OF SECURING NUT
D
E
B
B
A
H
C

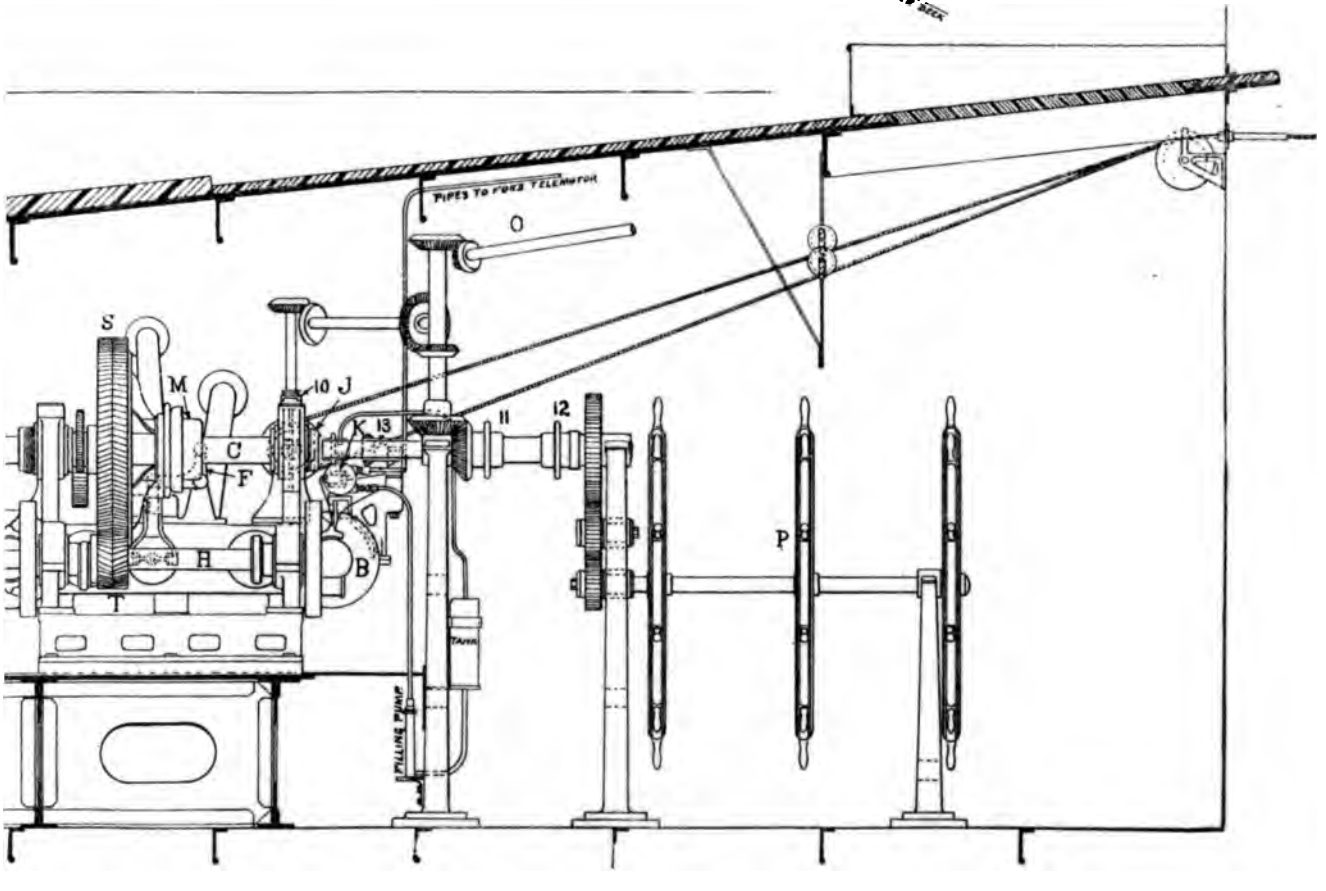

B
B
D
E
C


H
B
B
D
E
C

See Plates 35 and 36 for more recent arrangement of steering engine.




STEAM
EXHAUST
J
9
B
Q
R
F
E
7
8
13
S
M
K
11
12
C
H
P


PIPES TO FORE TELEMOTOR
O
S
M
10
J
K
13
11
12
C
F
H
B
T
P

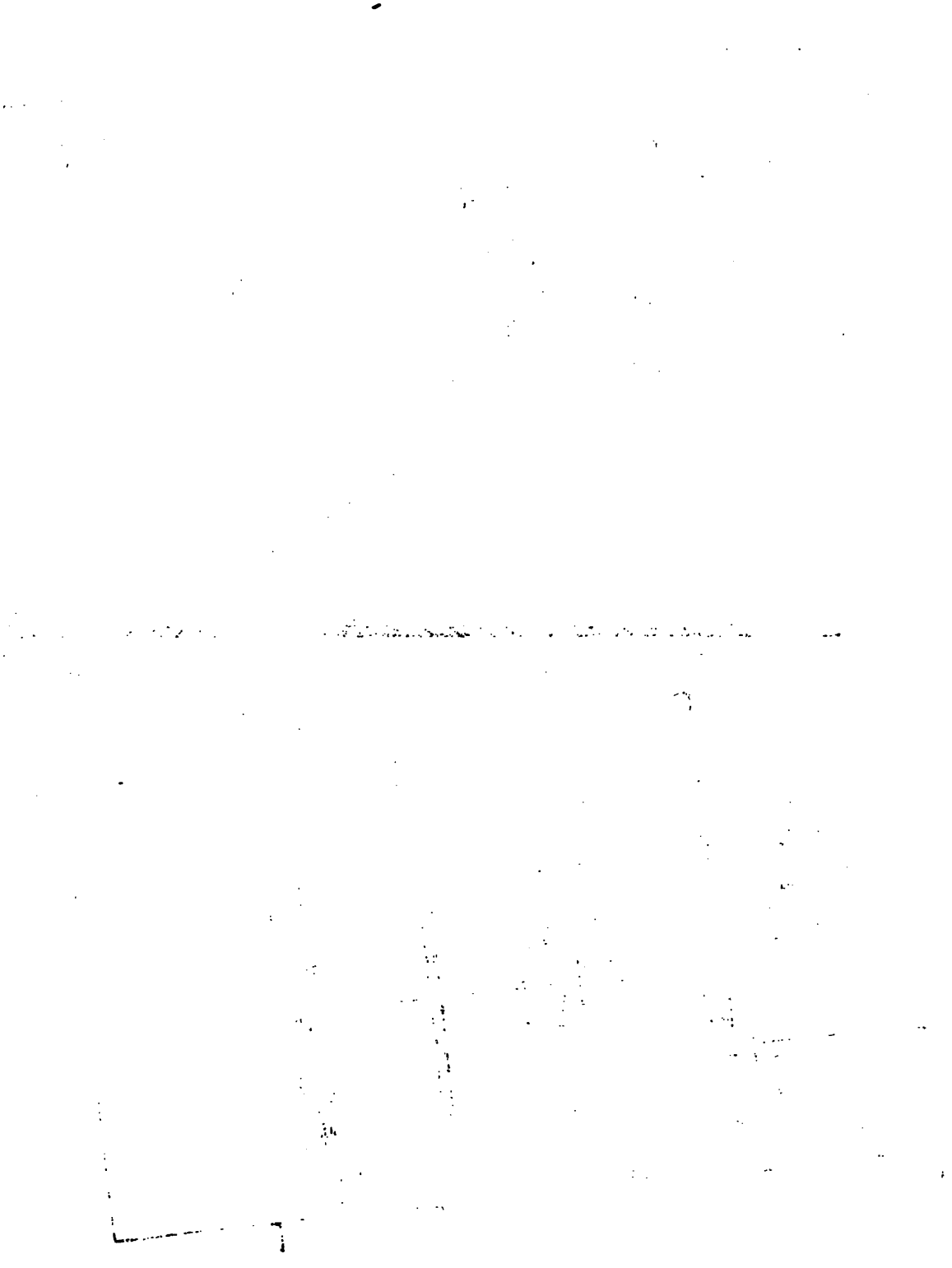
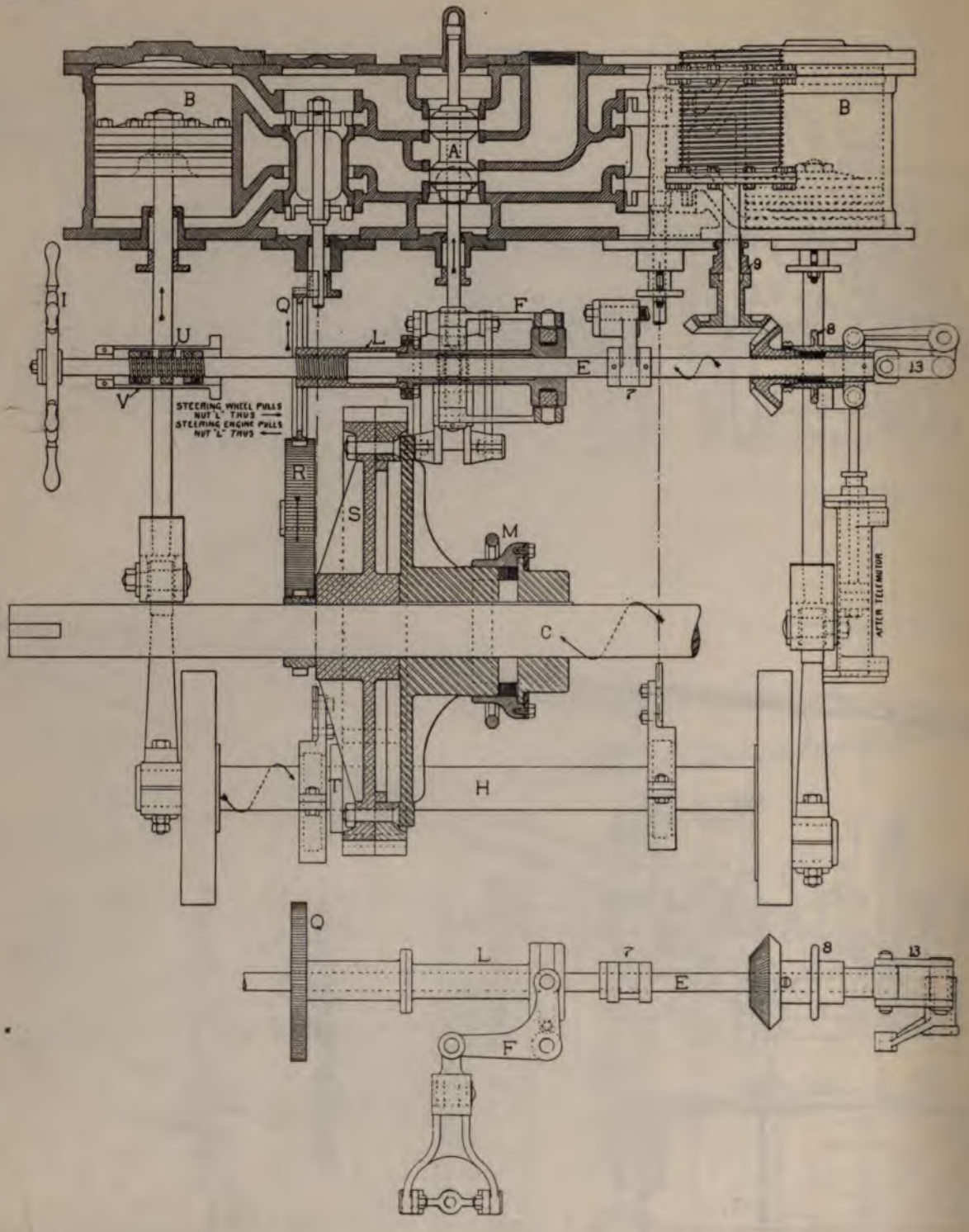
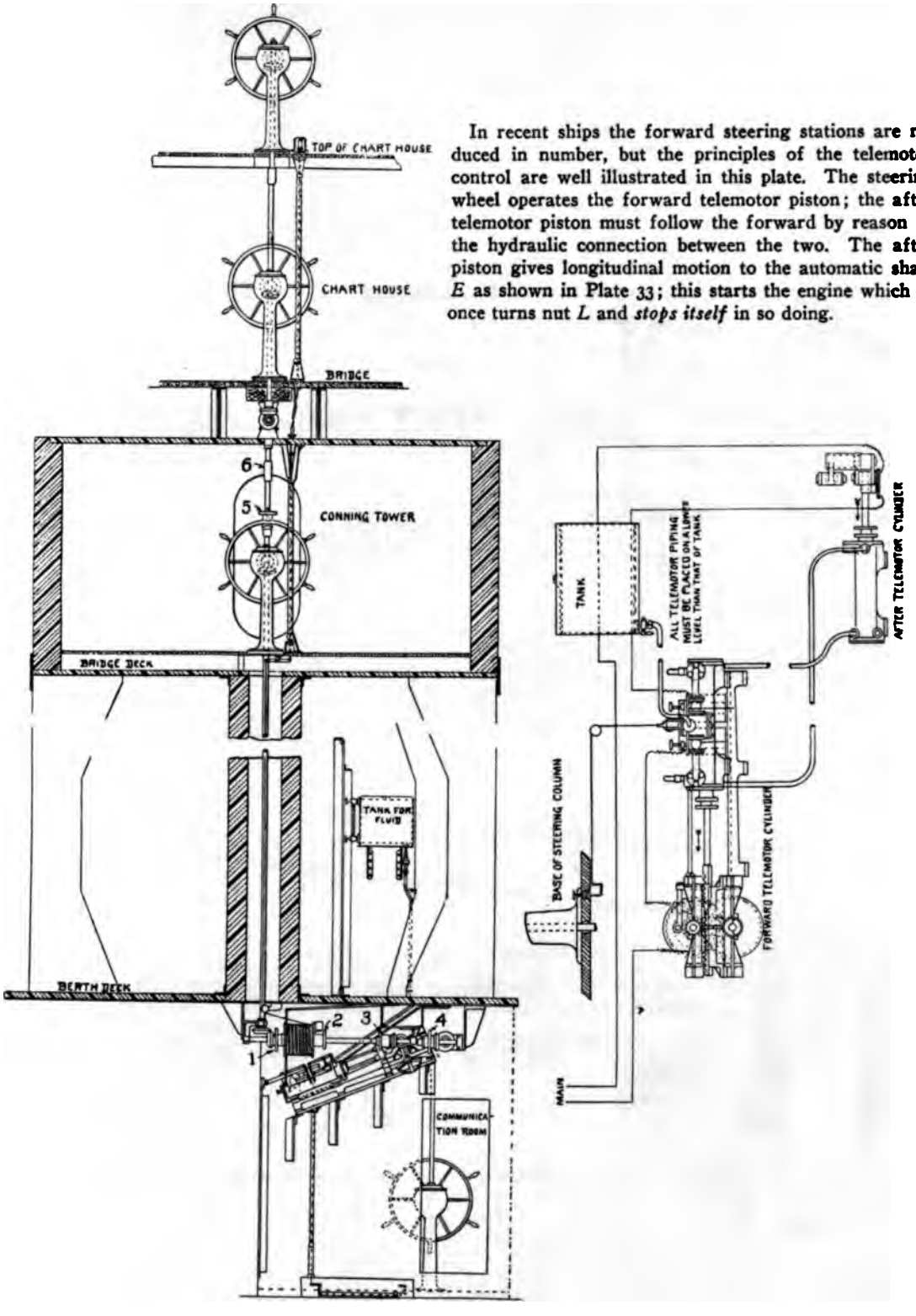


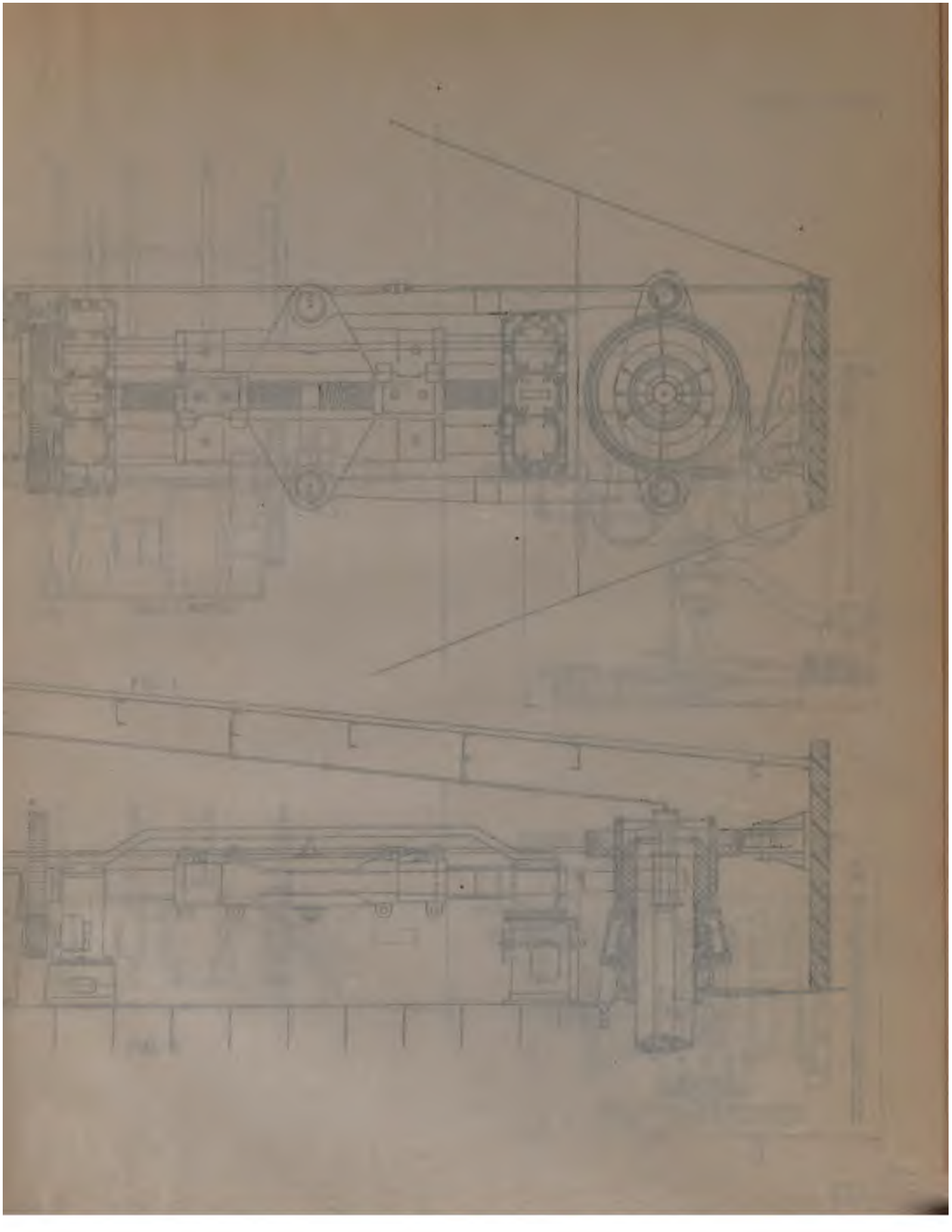
PLATE 32.

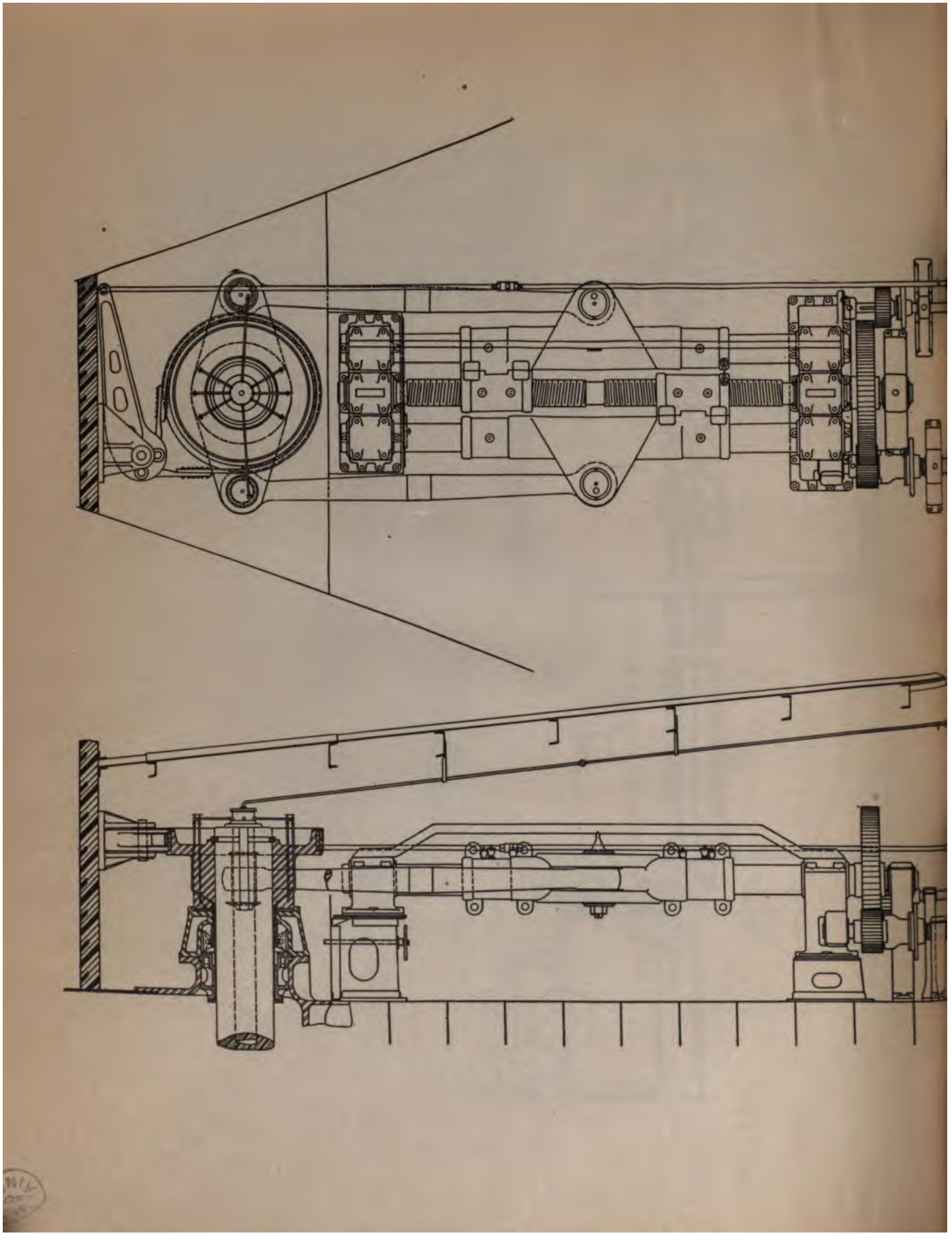
PLATE 33.

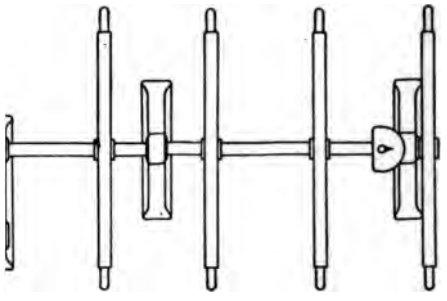

B
A
B
9
I
Q
F
U
L
8
E
13
V
7
STEERING WHEEL PULLS NUT 'L' THUS →
STEERING ENGINE PULLS NUT 'L' THUS ←
R
S
M
AFTER TELEMOTOR
C
T
H
Q
L
7
8
13
E
F



In recent ships the forward steering stations are reduced in number, but the principles of the telemotor control are well illustrated in this plate. The steering wheel operates the forward telemotor piston; the after telemotor piston must follow the forward by reason of the hydraulic connection between the two. The after piston gives longitudinal motion to the automatic shaft *E* as shown in Plate 33; this starts the engine which at once turns nut *L* and ***stops itself*** in so doing.







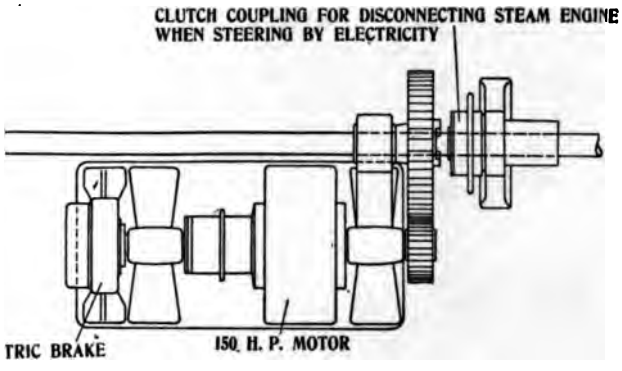
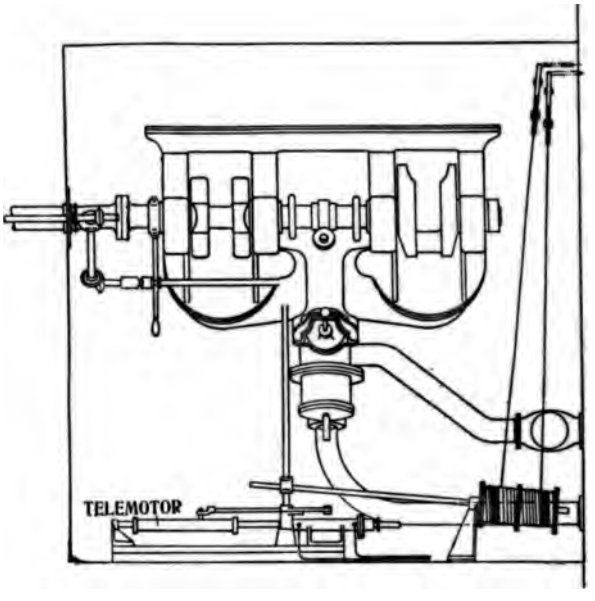


FIG. 1.



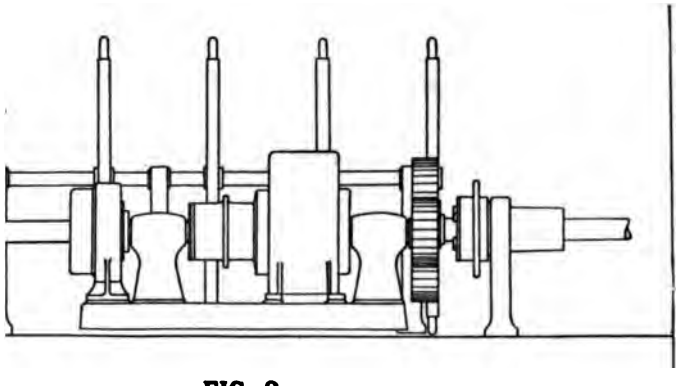
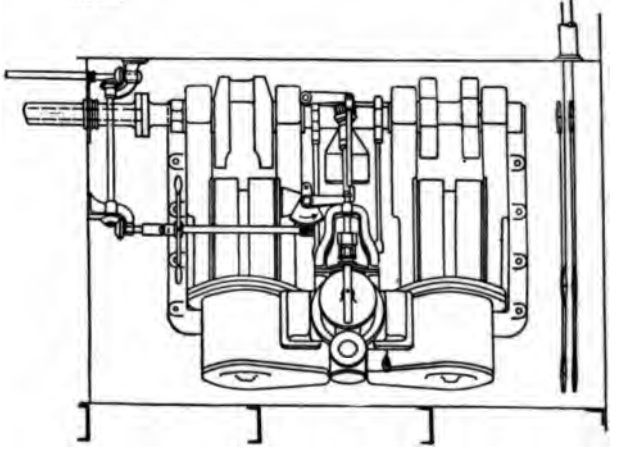


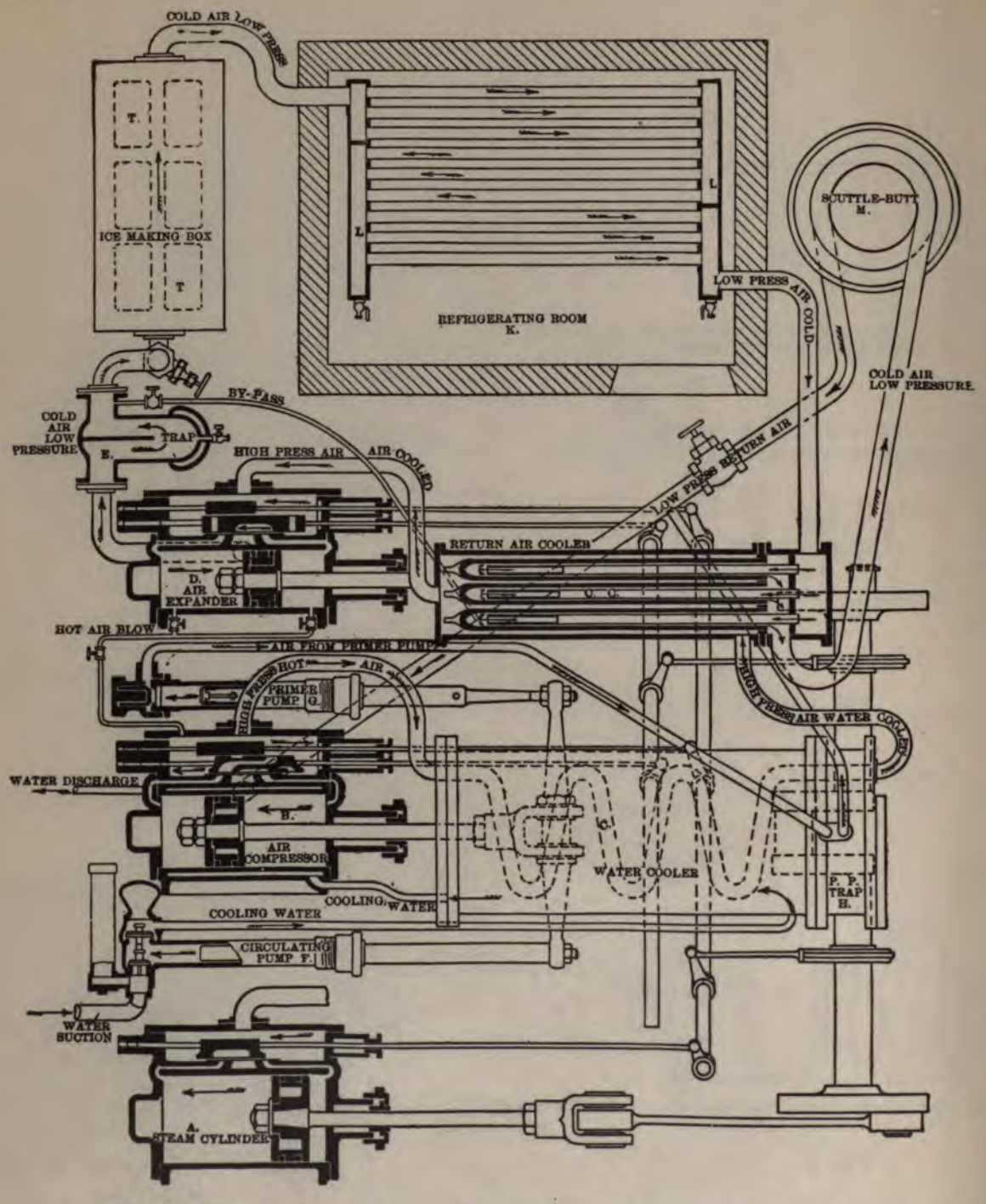
FIG. 2.

STEERING ENGINE LOCATED ADJACENT TO MAIN ENGINE ROOM



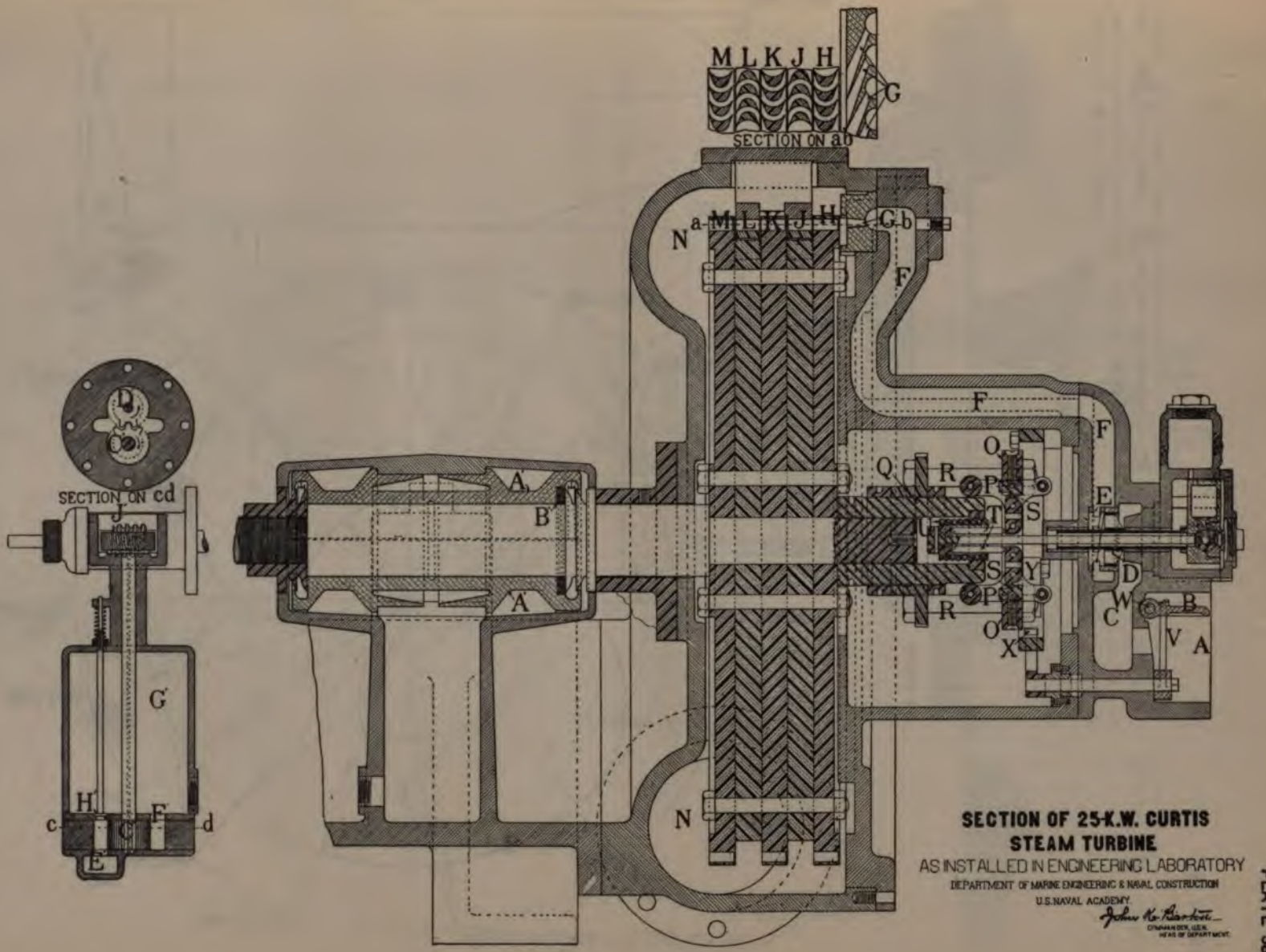
THE ALLEN DENSE AIR ICE MACHINE DIAGRAMMATIC SKETCH

PLATE 37.



THE ALLEN DENSE AIR ICE MACHINE DIAGRAMMATIC SKETCH

UNIV. OF MICH.



SECTION OF 25-K.W. CURTIS STEAM TURBINE

AS INSTALLED IN ENGINEERING LABORATORY

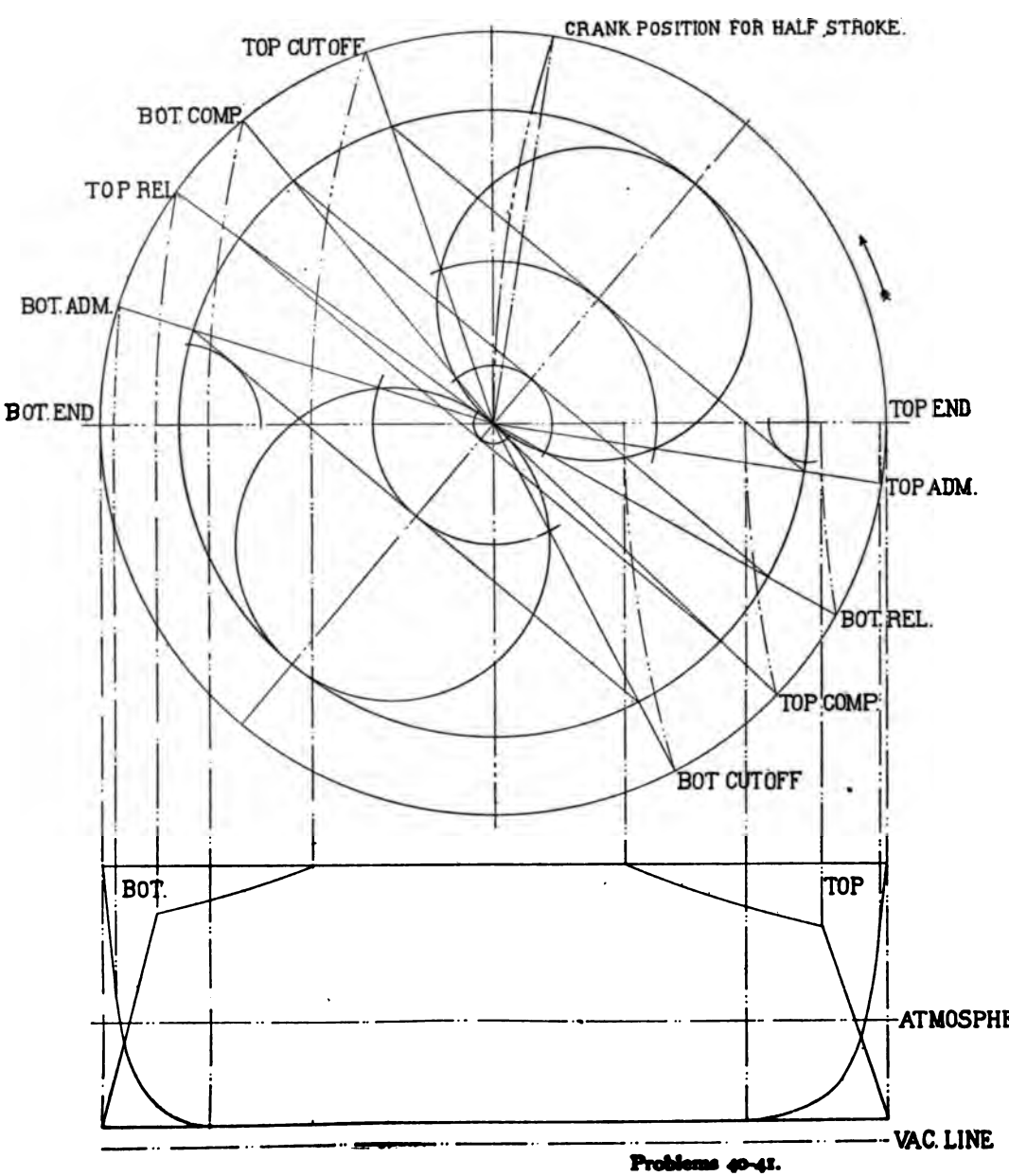
DEPARTMENT OF MARINE ENGINEERING & NAVAL CONSTRUCTION

U.S.NAVAL ACADEMY.

John R. Bartlett

COMMANDER, U.S.N.
HEAD OF DEPARTMENT.

PLATE 38.



Problems 40-41.

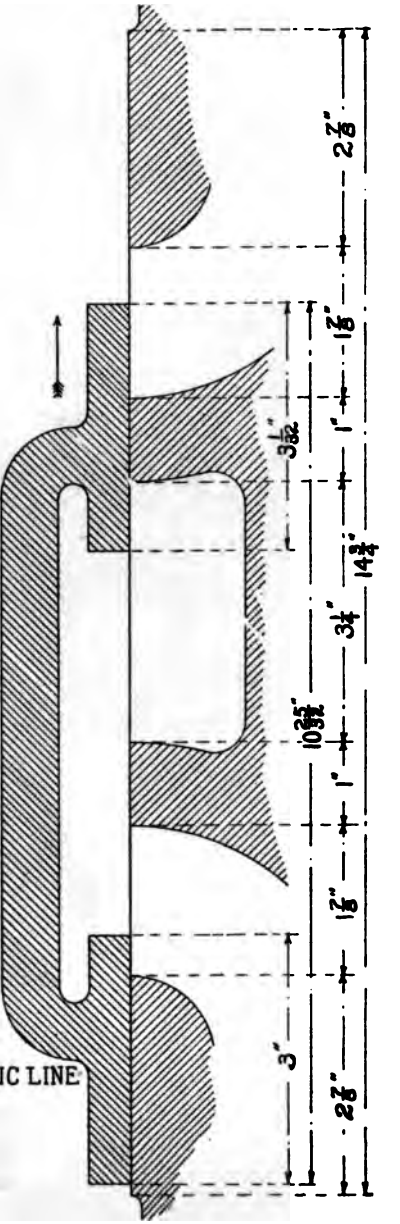
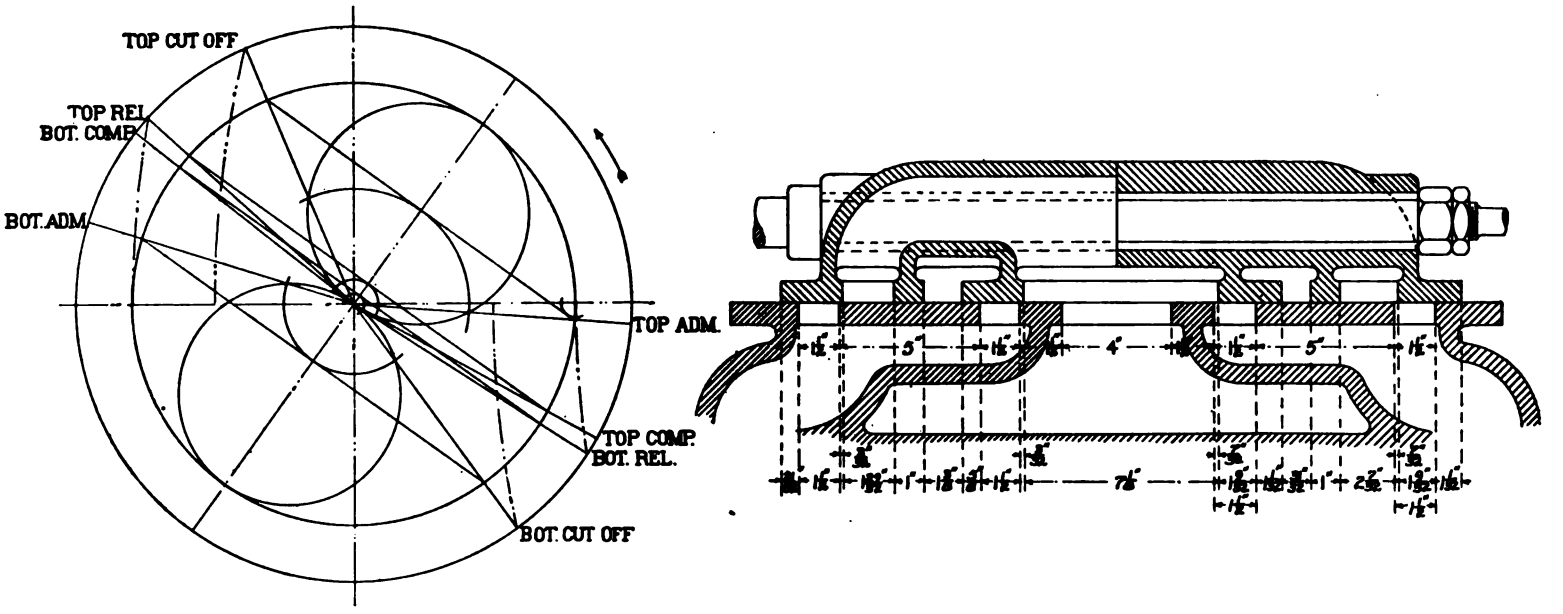
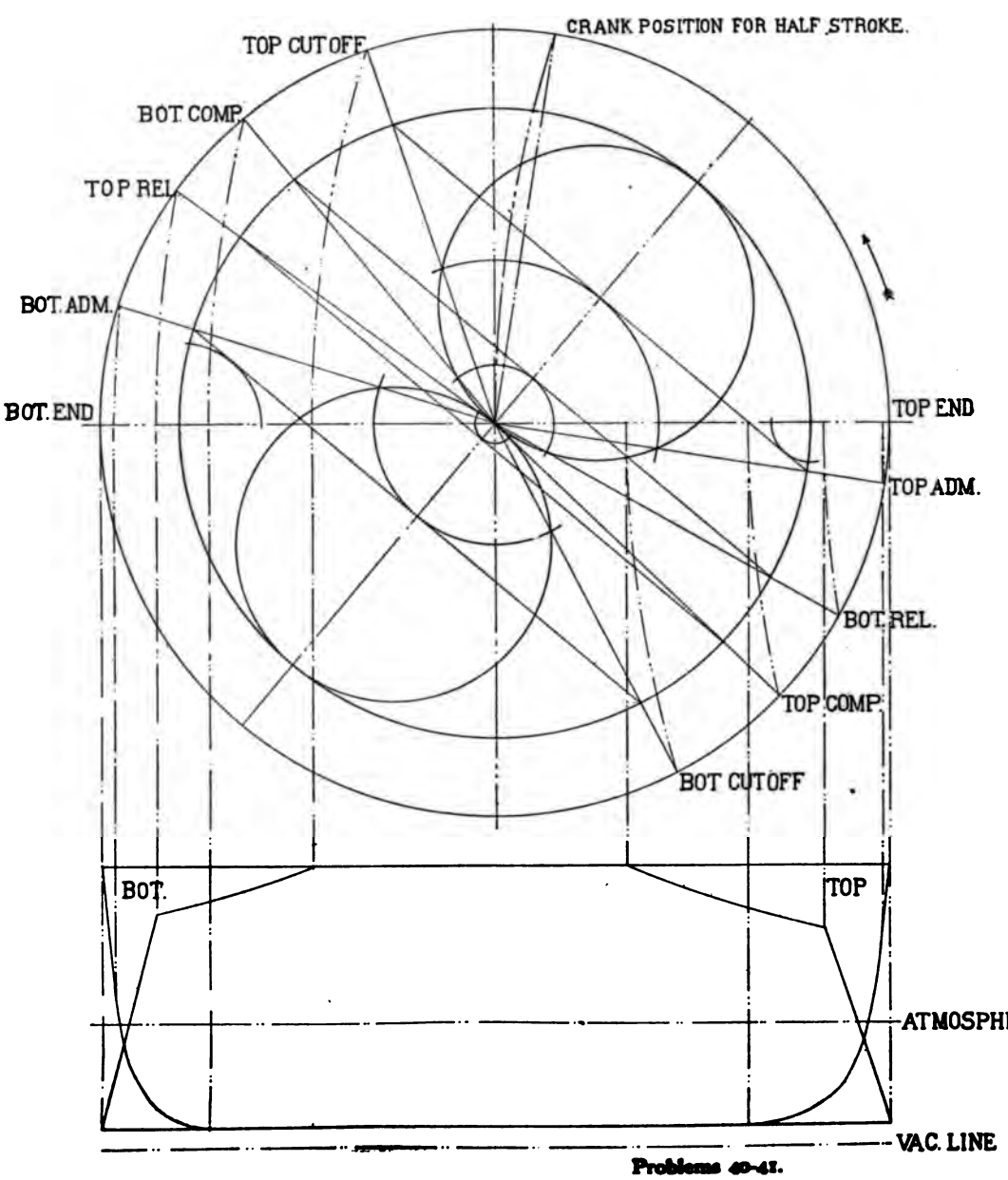


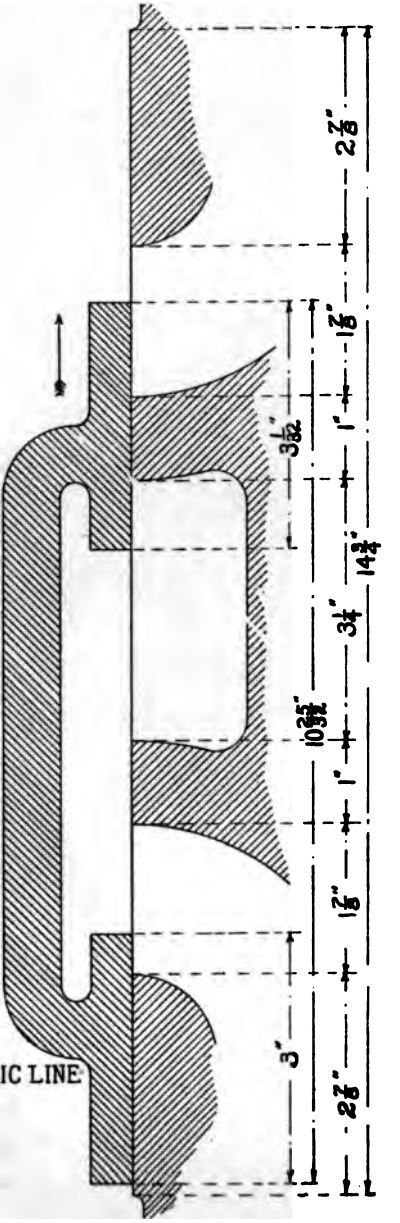
PLATE 39.



Problems 43-44.

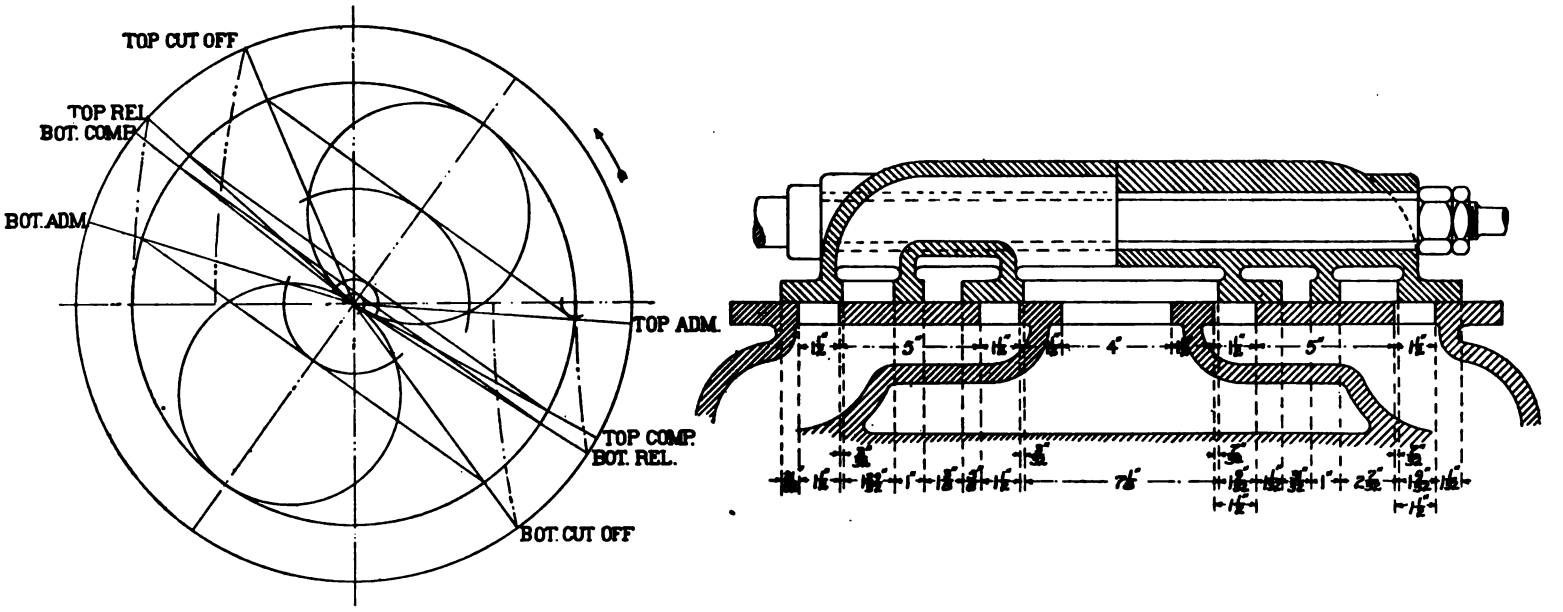
PLATE 40





Problems 40-41.

PLATE 99.



Problems 43-44.

PLATE 40

SCREW PROPELLERS.

METHOD OF DESIGN, DESIGN BY COMPARISON; EFFECT OF THRUST DEDUCTION AND REDUCTION OF DIAMETER ON THE PROPULSIVE COEFFICIENT.

By Captain C. W. Dyson, U. S. Navy.

INTRODUCTION.

In estimating the power required for the propulsion of any proposed hull, the Navy Department has practically abandoned all other methods than that of trials of the model of the proposed hull in the model tank, and from the results of these trials, obtains the curves of effective horsepowers based on speeds as abscissas. Other methods are used, however, as checks.

By the term "Effective Horsepower" is meant that power which when applied to a tow rope would pull the hull through the water at the desired speed.

The effective horsepower curves, as usually supplied to the Designing Branch of the Bureau, from which to estimate the necessary power for propulsion, are the curves of effective horsepower for the bare hull only. Before the estimation of power can be made, the expected increase of effective horsepower due to the resistance of the appendages, or excrescences, as they are sometimes called, must be added to the bare hull power curves to obtain the total effective horsepower necessary.

These appendages consist of the following, the various items being given in the order in which they most frequently occur:

1. Rudder and Stern Post.
2. Bilge Keels.
3. Struts, Bosses and Shafting.
4. Dockings Keels.
5. Small Scoops over openings in hull.
6. Large Scoops over openings in hull.

No. 1 is encountered in all vessels, either single or multiple screw.

No. 2 is met with in most vessels of any considerable size.

No. 3 exists only in vessels having two or more propellers, although in some cases of single-screw vessels, the dead wood may be cut away and the propeller shaft supported by a strut. In some cases of twin-screw ships, the form of stern known as the "Lundborg" stern may be used and there will be no struts.

No. 4 is only met with in large, heavy vessels where such keels are required to better distribute the weight of the hull when docking.

No. 5 is found in all torpedo boats and destroyers built at the present date.

No. 6 is found in those torpedo boats and destroyers built from ten to twelve years ago.

The resistances due to the bilge and docking keels and shafts are probably those due to their wetted surfaces only and can be calculated as such. The other appendages enter into the total residual resistance (wave and eddy making), and are estimated as varying according to Froude's Law of Comparison. That this law is correct when applied to such forms, is not certain, but it is adhered to for want of something more definite.

DERIVATION OF BLOCK COEFFICIENT TO USE IN CALCULATION OF PROPELLER.

Should we, for different classes of vessels, adopt standard sets of bow and stern lines and standard shapes of midship sections, we would have for all vessels of any class, no matter what the ratio of Beam to Length on the Load Water Line, an even condition of circumstances governing the flow of water to the propellers, and a change of percentage appendage resistance varying approximately directly as the ratio of Beam to this Length. We would find, however, that the nominal block coefficients ($= 35 \times$ Displacement $\div$ Beam $\times$ Length on Load Water Line $\times$ Draught), would change nearly inversely with the beam ratio, while the actual body conditions, so far as affecting the propeller performance, would have remained the same.

Further, should two models of equal beam, length and displacement, but of different midship section forms, be towed, it would be found that the model having the finer midship section coefficient ($=$ Area of midship section $\div$ Beam $\times$ Draught) would require the greater tow-rope power. It would also be found that the longitudinal coefficient of this model ($=35 \times$ Displacement $\div$ Area of midship section $\times$ Length of Load Water Line) would be the larger of the two.

For two vessels of the same dimensions and for the same service, the appendage resistances would be the same, but if one vessel had a finer midship section than the other, the bare hull resistance of the former vessel would be greater than the latter, therefore the percentage appendage resistance would be less.

As the corrective Chart for Block Coefficient and the Chart for estimation of Appendage Resistance are closely related, it will be well to explain them here and also explain how corrections for Block Coefficient are made when the location of the propellers differs from the location of the propellers on the Standard vessels whose propellers and results were used in the determination of the Charts of Propeller Design.

CHART 1.—FOR CORRECTION OF BLOCK COEFFICIENT.

On this Chart the abscissas are Block Coefficients, while for the Block Correction, the ordinates are values of Beam ÷ Length on L.W.L. For checking the Block by means of the Coefficient of Immersed Midship Section, the ordinates are Coefficients of Immersed Midship Section. All vessels, whose Immersed Midship Section coincides with the Standard Curve, will be of standard fore and after body (abnormal designs of hull not being considered). Those plotting below the Standard Curve will be bluffer, forward and aft, and those plotting above the curve will be finer than the Standard Hulls, so that for vessels that plot below the Standard Curve, the percentage appendage resistance will be less, and for those that plot above, the percentage appendage resistance will be greater than Standard by the amount shown by the curves.

USE OF CHART 1 IN PROPELLER DESIGN.

In the lower section of this Chart are shown three diagonal lines, *X*, *Y* and *Z*. Line *X* is for the Standard vessels from which the Charts of propeller design were obtained. Such vessels have coefficients of Immersed Midship Section falling close to the curve of M.S. coefficient marked Standard, have propellers located well clear of the hull so that loss, through interference of flow of water to the screw by the hull, is a minimum and also the effect of the wake on the velocity of the flow of water to the propeller is a minimum.

As the location of the screws draws closer in behind the hull and the influence of the wake has sensibly increased over that of condition *X*, the line *Y* takes the place of *X*. Where the propeller is located close to and directly to the rear of the stern post, so that the full wake effect of the hull is encountered, line *Z* replaces both *X* and *Y* in the determination of the Block Coefficient.

To apply this Chart, let us suppose we have three vessels, A, B and E, having the following characteristics:

	A.	B.	E.
Nominal Block Coef.....	.78	.665	.61
Beam ÷ Length on L.W.L.....	.123	.186	.217

It is required to find the Block Coefficient to use in the design of the propeller and also to estimate for the expected appendage resistances.

Plot *A*, *B* and *E* on the chart with the nominal block coefficients as abscissas and with the values of Beam ÷ L.L.W.L. as ordinates. Through these plotted points and the unity abscissa point, pass a straight line, extending it until it cuts line *X*. In the cases taken, *A*, *B* and *E* are all on the same line passing through the unity value of abscissas. Where this line crosses *X*, at *B*, project up to the Standard curve of midship section coefficient. Should the M.S. coefficient of the vessel in question plot

near to the standard curve of M.S. coefficient, the vessel's ends may be considered standard, and the vessel's block coefficient be taken as that given by the abscissa value of *B*. Should it fall above this curve, that is, the M.S. be fuller than standard, the ends will be finer; if below, fuller than standard, and the block coefficients be modified accordingly. Should the propellers be located in the conditions given by lines *Y* or *Z* the fineness will be gaged, as before, by the intersection of the line with *X*, but the actual block to use for the propeller design will be that given by the abscissa value of the point of intersection with *Y* or *Z*, except where correction is made for variation from the Standard M.S. Coefficient.

CHART 7.—CORRECTION FOR THRUST DEDUCTION.

Another point to be considered, in the estimate of power and the design of the propeller, is the peculiar effect upon thrust of the propeller exercised by fullness of after-body lines of the hull. This effect is not met with or is of such small extent as to be neglected, where the standard ship block coefficient to be used is .55 or less. As the standard ship block increases above .55, this evil effect becomes exaggerated and a corrective factor must be applied in designing the screw. This corrective factor is given by the Curves *C*, C^2 and C^3 on Chart 7, and is inserted in the equation for diameter, which is expressed thus:

$$\text{Diameter in feet} = \sqrt{\frac{2.88 \times \text{E.H.P.} \times C^3}{(\text{P.A.} \div \text{D.A.}) \times \text{E.T.} \times \text{S}}},$$

where E.H.P. = effective horsepower for the hull with appendages.

P.A. ÷ D.A. = Projected Area divided by Disc Area of screw = Projected Area Ratio.

S = Speed corresponding to the E.H.P.

E.T. = Effective thrust in lbs. per sq. in. on projected area of the propeller = $\dfrac{\text{E.H.P.} \times 33{,}000}{\text{S} \times 101.33 \times \text{Proj. Area in sq. in.}}$

101.33 = Speed in feet per minute for a speed of one knot per hour.

The effect of this corrective factor is to increase diameter and pitch and decrease revolutions of the propeller. In most of the cases where it becomes necessary to use *C*, the diameters of the propellers that can be used are limited to a certain maximum size; this allowable size is usually smaller than the theoretical figures call for, so that for the same projected area as the propeller of full diameter, the projected area ratio of the actual screw is much increased. Further, by the use of *C*, the thrusts per square inch of projected area are very much lower than those of the Propeller Design Charts for equal tip speeds.

This lowering of thrusts is a direct result of the excessive value of the wake and of the thrust deduction in such vessels, and no attempt has ever been made heretofore, so far as the writer is aware, to give a guide to the correction needed for these conditions of wake and thrust deduction.

The effect of reduction of diameter on propulsive coefficient will be shown later in discussing particular cases.

"THRUST DEDUCTION" ALSO CALLED "AUGMENT OF RESISTANCE."

A propeller driving a vessel acts very much after the nature of a pump, creating a suction column of water from forward to, and a pressure column of water aft from the screw. There is one difference in action to be noted, and that is the fact that the velocity of flow of the water in the suction column is less than it would be in a pump, due to the advance of the propeller itself along this column.

Should the propeller be so located in reference to the hull as to have the flow of water in this suction column seriously interfered with, so that the full supply of water which should flow, due to the suction head, is not given to the screw, a serious loss of propelling power will result.

The form of the hull, where the propeller is located in close proximity to it, may have such an adverse effect upon the screw as to make the obtaining of the designed speed an impossibility with the existing conditions. Such a case occurred a short time ago with two oil-fuel barges built for the Navy Department. These vessels were designed for a speed of six knots, and the effective horsepower curves indicated that this speed could be easily obtained with the power. However, after trying seven different propellers and developing a maximum I.H.P. of 176, the highest speed realized was only 5¼ knots.

The propeller had been located in the orthodox position during these trials, and the action of the water indicated that a portion of the feed was being drawn from astern. The line of shafting was then changed so as to lower the propeller about three feet, the lower blade projecting below the line of keel. In this new position, with propellers of the same pitch and surface but of six inches greater diameter, a speed of 6¼ knots was obtained with the same power. The greater portion of this increase in efficiency may be credited to the increase in diameter, and the remainder to the change in position.

Barnaby, in writing on this subject, makes the following statement: "If a screw is placed behind a stern so bluff that the supply of water is impeded, it will draw in water at the center of the driving face and throw it off from the tips of the blades like a centrifugal pump. It is recorded that an attempt to propel a square-ended caisson by means of a screw resulted in the caisson going astern, whichever way the screw was driven."

CHART 2.—ESTIMATE OF APPENDAGE RESISTANCE.

This subject is one of which little is known, and yet it is one of vital importance in the design of the propeller and the estimate of power required for propulsion, if anything like approximately accurate charts for design purposes are to be constructed.

For standard forms of forward and after bodies of the hull the writer has assumed that the appendage resistance percentages will vary directly as the ratio $\frac{B}{L.W.L.}$. Any error entering into this assumption is on the right side of the ledger and will result in a slightly higher estimate of resistance than actually exists.

Should the hull be finer than the standard the percentage resistance will increase above that given by Chart 2; should the hull be bluffer than standard the opposite effect will be produced, the amount of increase or decrease being indicated on Chart 1, curves of midship section coefficient.

On Chart 2 are drawn, with $\frac{B}{L.W.L.}$ as abscissas and values of percentage appendage resistance as ordinates, lines of percentage resistance for each appendage in the order in which they usually occur. Thus, the lowest line is for the rudder and rudder post, which are found on all vessels. The next line above, No. 2, is for the rudder, rudder post and bilge keels; No. 5 is for the rudder, rudder post, bilge keels, shafts, bosses and struts, when carefully designed and placed: and finally, the line for total appendage resistance, No. 7, which, in addition to the appendages covered by No. 5, includes docking keels.

On the Chart is given a table of conditions governing the lines of resistance shown.

Should the model be towed with any of the appendages the percentage resistance due to these appendages should be subtracted from the total appendage resistance given by Chart 2 for all the appendages which the vessel under consideration is to carry, and the remaining percentage be applied to the E.H.P. given by the model-tank curve.

The usual method of judging of the performance of a propeller is to obtain the ratio $\frac{E.H.P.}{I.H.P.}$ or $\frac{E.H.P.}{S.H.P.}$, and the higher these ratios the better are the propellers. Very often, and in fact more often than otherwise, the E.H.P. used in either of these ratios is for the bare hull only. This comparison, when the E.H.P. of the bare hull is used, is not just, and many a good propeller has been rated as a poor one on account of it. The real E.H.P. delivered by the propeller is that necessary for the hull with all appendages, and this is what should be used in estimating the propeller performance.

In order to better illustrate the handling of Charts 1, 2 and 7 the following two problems are submitted:

Problem (*a*).

Vessel having the ratio $\frac{B}{L.L.W.L.} = .200$;

Block Coefficient $= \frac{\text{Displacement} \times 35}{\text{L.L.W.L.} \times \text{Beam} \times \text{Draught}} = .525$;

Coefficient Immersed Midship Section $= .975$.

a. Twin Screws: (Appendages—Rudder, rudder post, shafts, struts and bosses, bilge keels.)

b. Single Screw: (Appendages—Rudder, rudder post and bilge keels.)

Find block coefficient for each case for use in propeller design, percentage appendage resistance and the thrust deduction factor *C*.

Solution.—Plot the position of the vessel on Chart 1, with the B.C. $= .525$ as abscissa and $\frac{B}{L.L.W.L.} = .2$ as ordinate, and through this plotted point and through the abscissa value of unity at *D* draw a straight line. This line crosses the standard line of B.C. at an abscissa value of .6, and the standard value of midship section coefficient corresponding to this abscissa value of .6 is .975, the same as that of the vessel; therefore the vessel's lines are regarded as standard. The value of B.C., therefore, for the twin-screw condition is .6, and for the single-screw condition will be at the intersection of the line drawn through the plotted position of the vessel and the line *Z*, for which the abscissa value $= .72$

As the lines are standard no further correction than the above is necessary, and we take from Chart 2, line 5, abscissa value .2, the percentage correction for the appendages $= 19.6$ per cent. for the twin-screw arrangement, while for the single-screw arrangement this percentage $= 4.6$ per cent. is taken from Chart 2, line 2. For the thrust deduction factor *C* we enter Chart 7 with an abscissa value of .6 for which $C = 1.04$ for the twin-screw, and an abscissa value of .72 for which $C = 1.23$ for the single-screw arrangement.

Problem (*b*).—Vessel of the same ratio $\frac{B}{L.L.W.L.} = .200$; same B.C. value $= .525$, but having a coefficient of immersed midship section .9. Two conditions as before, namely, twin and single screws with the same appendages as before. Find B.C., appendage percentage resistance and *C*.

Solution.—Plot the vessel's position on Chart 1 and draw the line through *D* as before. This line will intersect *X* and *Z* at abscissa values of .6 and .72 as before, but these are not the true block coefficients to use in obtaining slip values, as the standard midship section coefficient corresponding to abscissa value of .6 is .975, while the vessel in question has a M.S. coefficient of .9. Assuming that the fullness of the

vessel's lines will vary inversely as the M.S. coefficient we have Slip B.C. = Standard B.C. $\times \frac{\text{Standard M.S. Coefficient}}{\text{Actual M.S. Coefficient}}$; $.6 \times \frac{.975}{.9} = .65$ for the twin-screw arrangement.

Drawing a line through the point on X having an abscissa value of .65, and through D, we find the intersection of this line with Z to have an abscissa value of .76, and this latter value will be the slip B.C. to use for the single screw. Referring again to Chart 1, Curves of M.S. coefficient, we see that for an abscissa value of .6 and M.S. coefficient of .9 the appendage resistance percentage is only 80 per cent. of the standard resistance given by Chart 2. By Problem 1 this was found to be 19.6 per cent. for the twin-screw and 4.6 per cent. for the single-screw arrangement; therefore, for the vessel of Problem 2 these percentages become $19.6 \times .80 = 15.68$ per cent. and $4.6 \times .80 = 3.68$ per cent. For the thrust deduction factors C we enter Chart 7 with abscissa values of .65 and .76 for the two conditions and the values obtained are for B.C. = .65, $C = 1.1$, and for B.C. = .76, $C = 1.31$.

The Screw Propeller.

THEORIES OF DESIGN.

The three most important theories of design, given in the order of their importance, are: 1. Froude's; 2. Rankine's, and 3. Greenhill's. The assumptions for each of these being as follows:

Froude.—Assumes the element as a small plane moving through the water along a line which makes a small angle with the direction of the plane. He then takes the normal pressure upon the elementary area, which gives propulsive effect to vary as the area, as the square of its speed and as the sine of the slip angle.

Rankine.—The fundamental assumption was that as the propeller advanced with a certain slip all the water in an elementary ring of radius r was given a certain velocity in a direction perpendicular to the face of the blade at that radius. Then, from the principle of momentum, the thrust from the elementary ring is proportional to the quantity of water acted upon in one second, and to the sternward velocity communicated to it.

Greenhill.—Approaches the problem from a direction entirely different from that of either of the other two preceding theorists. He assumes that the propeller is working in a fixed tube with closed end. The result is that the motion transmitted to the water is wholly transverse. The blade is assumed perfectly smooth, so that the pressure produced by the reaction of the water is normal to the blade and has a fore-and-aft component which produces thrust.

In all the above theories, the loss by friction is taken as that due to the friction of the propelling plane moving edgewise, or nearly so, through the water.

In all of the theories in connection with which mathematical methods are to be used, it is practically necessary to regard the blade as having no thickness. This is a serious defect in all the theories, as they all use a true slip based upon true pitch and consider the designed pitch of the driving surface of the blade as this true pitch. The fact of the matter is that the face pitch of a blade with thickness, or its nominal pitch as it may be called, is very different from the true or actual pitch, and this fact causes complications in using the mathematical formulas.

It would be necessary, in case these theoretical formulas were adhered to, to compare each formula with experimental results and select that one which seemed to agree most closely. Then, using this as a semi-empirical formula, with coefficients and constants deduced from experiments or experience, problems could be satisfactorily dealt with. When the vast number of various conditions for which we may be called upon to design a propeller are considered, it is readily seen how impossible it would be to tabulate the correctors which would be required to cover all, or even a large number of these conditions.

Casting aside the idea of designing propellers from purely theoretical formulas, as incorrect and impractical, there are then remaining for use the following

PRACTICAL METHODS OF DESIGN.

1. By "Direct Comparison," when all the conditions for a satisfactory vessel of a similar form to the one under consideration are known.

2. By methods based on trials of model propellers in model tanks. This may also be classed under the head of "direct comparison," as the Laws of Comparison are assumed to cover propellers as well as hulls.

3. By methods based on actual trials of full-sized propellers in service over carefully measured courses.

The first method practically insures a propeller of equal propulsive efficiency with that of the propeller on the compared vessel, but gives no opportunity for improvement in performance.

The second method is open to the decided objection that the conditions under which the model screw is tried in the tank are radically different from those under which the full-sized screw operates. In fact, propellers whose models have shown high tank efficiencies, have failed most signally in service, while other propellers whose models gave poor efficiency, have delivered a high propulsive coefficient. This latter has been ascribed to a high hull efficiency, but this explanation does not exactly satisfy when the fact is considered that where two or more propellers for the same vessel have been tested, that propeller whose model gave the highest efficiency has failed, while the propeller with the lower tank efficiency has succeeded.

The writer is inclined to the belief that the true cause of these discrepancies exists in the use of an incorrect method of derivation of the model screw dimensions from those of the full-size propeller.

METHOD OF "DESIGN BY COMPARISON."

The following orthodox method is taken from Peabody's Naval Architecture, and is that which is generally used:

D_1 = Diameter of original propeller.
D_2 = Diameter of 2d propeller or of model propeller.
L_1 = Length of compared vessel.
L_2 = Length of new vessel or model hull behind which model screw should operate if fitted to a hull (which is not done).
P_1 = Pitch of original propeller.
P_2 = Pitch of 2d or model propeller.
R_1 = Revolutions of original propeller.
R_2 = Revolutions of 2d or model propeller.
S_1 = Speed of compared vessels.
S_2 = Corresponding speed of new or model hull.

Then

$$D_2 = D_1 \times \frac{L_2}{L_1} = D_1 r;$$

$$R_2 = R_1 \left(\frac{L_1}{L_2}\right)^{\frac{1}{2}} = \frac{R_1}{r^{\frac{1}{2}}};$$

$$S_2 = S_1 \left(\frac{L_2}{L_1}\right) = S_1 r^{\frac{1}{2}};$$

$$P_2 = pP_1 = rP_1$$

where same ratio of $\frac{P}{D}$ is retained for the model as that of the original propeller.

$$\text{Apparent slip}_1 = \frac{P_1 \times R_1 - 101.33\, S_1}{P_1 \times R_1}.$$

$$\text{Apparent slip}_2 = \frac{pP_1 \times \frac{R_1}{r^{\frac{1}{2}}} - 101.33\, S_1 r^{\frac{1}{2}}}{pP_1 \times \frac{R_1}{r^{\frac{1}{2}}}} = \frac{pP_1 \times R_1 - 101.33\, S_1 r}{pP_1 \times R_1};$$

which, when $p = r$ becomes $\frac{P_1 \times R_1 - 101.33\, S_1}{P_1 \times R_1}$, or the apparent slip of the model, is equal to the apparent slip of the original propeller.

$\text{Tip speed}_1 = R_1 \times \pi D_1$;

$\text{Tip speed}_2 = R_2 \times \pi D_2 = \frac{R_1}{r^{\frac{1}{2}}} \times \pi D_1 r = R_1 \pi D_1 r^{\frac{1}{2}}$; and

$\text{T.S.}_2 \gtrless \text{T.S.}_1$ depending upon the value of r.

Again,

$$\text{I.H.P.}_2 = \text{I.H.P.}_1 r^{\frac{1}{2}};$$

$$\text{Disc area}_2 = \tfrac{1}{4}\pi \text{D}_1^2 r^2;$$

$$\text{Disc area}_1 = \tfrac{1}{4}\pi \text{D}_1^2;$$

$$\text{P}_2 \times \text{R}_2 = \frac{p\text{P}_1 \times \text{R}_1}{r^{\frac{1}{2}}};$$

$$\text{I.T. per sq. in. Disc Area}_2 = \frac{\text{I.H.P.}_1 \times r^{\frac{1}{2}} \times 33{,}000}{\frac{p\text{P}_1 \times \text{R}_1}{r^{\frac{1}{2}}} \times \frac{\pi}{4}\text{D}_1^2 r^2} = \frac{\text{I.H.P.}_1 \times 132{,}000}{p\text{P}_1 \times \text{R}_1 \times \pi \text{D}_1^2};$$

$$\text{I.T. per sq. in. Disc Area}_1 = \frac{\text{I.H.P}_1 \times 33{,}000}{\text{P}_1 \times \text{R}_1 \times \frac{\pi}{4}\text{D}_1^2} = \frac{\text{I.H.P.}_1 \times 132{,}000}{\text{P}_1 \times \text{R}_1 \times \pi \text{D}_1^2};$$

$$\therefore \frac{\text{I.T.}_2}{\text{I.T.}_1} = \frac{r^2}{p}, \text{ or where } p = r, \frac{\text{I.T.}_2}{\text{I.T.}_1} = r.$$

In other words, with different percentage losses from blade friction due to change in tip speeds, the model screw is supposed to deliver an equal percentage of the power driving it as effective thrust, with the original propeller, and its apparent slip is supposed to be equal to that of the original propeller, although the thrusts per square inch of disc area have been changed in the ratio r, the two screws working under approximately the same conditions of resistance.

In the method of comparison proposed by the writer, and which method, by the way, may be considered as highly unorthodox, the formulas take the following forms:

$$\text{D}_2 = \text{D}_1 r^{\frac{1}{2}};$$

$$\text{P}_2 = \text{P}_1 r^2;$$

$$\text{R}_2 = \frac{\text{R}_1}{r^{\frac{1}{2}}};$$

$$\text{S}_2 = \text{S}_1 r^{\frac{1}{2}};$$

$$\text{Apparent slip}_1 = \frac{\text{P}_1 \times \text{R}_1 - 101.33\ \text{S}_1}{\text{P}_1 \times \text{R}_1};$$

$$\text{Apparent slip}_2 = \frac{\text{P}_1 r^2 \times \frac{\text{R}_1}{r^{\frac{1}{2}}} - 101.33\ \text{S}_1 r^{\frac{1}{2}}}{\text{P}_1 r^2 \times \frac{\text{R}_1}{r^{\frac{1}{2}}}} = \frac{\text{P}_1 \times \text{R}_1 - 101.33\ \text{S}_2}{\text{P}_1 \times \text{R}_1}.$$

$\therefore$ Apparent slip$_1$ = Apparent slip$_2$;

$$\text{Tip Speed}_1 = \text{R}_1 \times \pi \text{D}_1;$$

$$\text{Tip Speed}_2 = \text{R}_2 \times \pi \text{D}_2 = \frac{\text{R}_1}{r^{\frac{1}{2}}} \times \pi \text{D}_1 r^{\frac{1}{2}} = \text{R}_1 \times \pi \text{D}_1.$$

$\therefore$ Tip Speed$_1$ = Tip Speed$_2$.

Again,

$$\text{I.H.P.}_2 = \text{I.H.P.}_1 r^{\frac{1}{2}};$$

$$\text{Disc Area}_2 = \frac{\pi}{4} D_1^2 r^3;$$

$$\text{Disc Area}_1 = \frac{\pi}{4} D_1^2;$$

$$P_2 \times R_2 = P_1 r^2 \times \frac{R_1}{r^{\frac{1}{2}}} = P_1 r^{\frac{1}{2}} \times R_1;$$

$$\text{I.T. per sq. in. Disc Area}_2 = \frac{\text{I.H.P.}_1 \times r^{\frac{1}{2}} \times 33{,}000}{P_1 r^{\frac{1}{2}} \times R_1 \times \frac{\pi}{4} D_1^2 r^3} = \frac{\text{I.H.P.}_1 \times 132{,}000}{P_1 \times R_1 \times \pi D_1^2};$$

$$\text{I.T. per sq. in. Disc Area}_1 = \frac{\text{I.H.P.}_1 \times 132{,}000}{P_1 \times R_1 \times \pi D_1^2}.$$

$\therefore$ I.T.$_2$ = I.T.$_1$, and for the model screw the tip speed, apparent slip, and thrust per square inch of disc area are identical with those of the original screw. This identity existing in this case and not in the former, why should the latter not be taken as the proper method of comparison?

The method by which the latter method of comparison is derived will be explained later on.

THE THIRD "METHOD OF PRACTICAL DESIGN."

This method is open to the criticism that results are so easily affected by inaccuracies in timing runs over the measured course, by inaccurate count of revolutions and inaccurate power measurements, by variation of conditions as to smoothness of immersed body of the ship and surfaces of the propellers, variations in trim and displacement of the vessel from the model conditions, adverse conditions of wind and sea, and variable skill of helmsmen.

These variables may, with proper care, be reduced so as to have the least possible effect, and, when this care is taken, the results obtained are quite as good for designing purposes, if not better, than can be obtained by any other method.

In conformity with the foregoing statement, the writer has developed a series of Charts for use in propeller design, which, up to date, have given uniformly good results when basic hull data of sufficient accuracy has been at hand. A brief description of the development of the method and of the charts will now be given.

DEVELOPMENT OF THE METHOD.

In Froude's investigations of the action of screw propellers and in Barnaby's development of Froude's results, the model propellers used had a constant form of developed area of blade and a constant ratio of greatest width of development to the diameter of the screw.

The form of developed area used was an ellipse of which the major axis was the radius of the propeller, and this form was used for all values of pitch ratio (P÷D) that were experimented with.

The constants obtained by Barnaby are expressed by the following formulas:

$$C_A = \frac{A.V.^2}{I.H.P.};$$

$$C_R = \frac{R \times D}{V}, \text{ in which}$$

A = Disc area of the propeller (including hub), in square feet.
R = Revolutions per minute of the propeller.
D = Diameter of the propeller, in feet.
I.H.P. = Indicated horsepower applied on the screw = 2 × effective horsepower or tow-rope horsepower required for the bare hull at the speed S.
V = Speed of the water through the screw, in knots per hour.

For convenience in computation, in the above formulas, the speed of the water through the propeller was replaced by the speed S of the ship in knots per hour, an assumed standard wake of ten per cent. being applied. A standard thrust deduction of ten per cent. was also adopted, as representing the addition to tow-rope resistance caused by the action of the propeller on the stream lines of the ship.

Assuming that for propellers identical in all respects except that of developed area, the thrust that can be delivered will vary directly as the developed area, Barnaby's constants can be represented graphically as a chart, and on this chart can be plotted from their performances, the location of any desired propeller.

In developing the writer's method of design, many propellers were so plotted, and from the results were obtained a sufficient number of factors to enable those factors of propeller design, which directly affect the efficiency, to be so tied together graphically, as to render it possible to take from these charts, when the resistance and form of hull are known, the necessary factors for use in the determination of the diameter, pitch, projected area, and probable propulsive coefficient that will be obtained, for any desired case.

These charts are shown marked 3, 4, 5, 6 and 7, and the method of using them will be explained later.

STANDARD FORMS OF PROJECTED AREAS OF BLADES FOR USE WITH THE CHARTS OF DESIGN.

Returning to the Barnaby presentation of Froude's results, the constants obtained by Barnaby are only correct so long as the ratio of developed area to disc area and the elliptical form of this developed area used by him, are adhered to. There is no

way of allowing for the effect of increase or decrease in this developed area except the rough one of estimating that the total thrust that can be delivered by the propeller will vary directly as the developed area.

By investigation of what occurs when the standard elliptical blade used by Barnaby in his experiments is broadened or narrowed, it is readily seen that this method of correcting for variations in developed area ratio is incorrect, for as the blade widens, for any one pitch ratio, the length of its resistance arm increases above that of the standard width blade, and as it narrows, the length of this arm decreases. In the first case, the resistance of the blade to turning is increased not only by the increased surface friction of the larger blade, but also by the increase in the length of the radius to the center of pressure of the blade. Should the blade be narrowed below standard, the opposite effect will be produced.

If elliptical blades of varying pitch ratios, but having the same area of projection on the disc, be laid down, it will be noted that the form of projection, not only on the disc, but also on the plane of the axis of the propeller, changes as we pass from the lower to the higher pitch ratios. This same change in form of projected area also occurs if we lay down blades of the same pitch ratio but of different values of developed area ratio, thus in both cases showing that not only the resistance of the blade due to change of surface, and in the first case change of pitch, has been affected, but it has been still further modified by the change in the distribution of this surface and the modification in the leverage arm of the blade resistance.

A distinguished educator who, until lately, was an advocate for the use of constant developed area form in design work, has lately put before the public a work on propellers in which he advocates the use of the projected area in place of developed area. He has adopted, instead of the constant elliptical form of development, a constant elliptical form of projected area. Should the projected area ratio be .3 its form is that of an ellipse, and should this ratio be .6 the form is again an ellipse. The same changes in distribution of surface occur as before, and no benefit has been obtained except that of having an easy form to lay down and also one that can be mathematically represented.

In order to maintain as constant a distribution of blade area as possible and thus guard against changes in resistance due to changes in distribution the writer adopted as a basic form of projected area the form of projection of the blades of two propellers having approximately a projected area ratio of .32. These two propellers had most excellent records, which we could hardly hope to better.

Using a standard hub diameter equal to .2 the diameter of the propeller, this standard basic form of projection was drawn. Then, with the center of the hub as a center, and different radii, circular arcs were struck crossing the axis of the projection. In obtaining the projected forms for areas differing from the basic area the

widths of the projections measured on these circular arcs were made proportional to the circular arc widths of the basic projection; that is, a .6 projected area ratio would have circular arc measurements $\frac{60}{32}$ times as great as those for the basic .32 projection.

The forms of projected area so obtained, when compared with the forms of blades of many propellers, are found to agree very closely, from the lowest to the highest values of P.A.÷D.A., with those forms which have the best records credited to them.

By using these forms, for any pitch ratio the resistance arm of the blade always remains the same, no matter what the developed or projected area ratio, and the only change in resistance to turning is that caused by the additional surface. The forms of projection, both on the disc and on the fore-and-aft plane, remain constant for all pitch ratios.

Naval Constructor D. W. Taylor, in writing on the effect of blade form as deduced from model tank experiments, stated "A good practical rule would seem to be to make the blades broader at the tip for low pitch ratios and narrow them for high ones." The blades here advocated follow this rule automatically, as the broadest part of the developed blade, measured on the elliptical development of the circular projected arc, moves slowly in towards the hub as the pitch ratio increases, thus gradually narrowing the tip of the blade for the higher pitch ratios.

The derived projected forms are shown on Plate 48 (Chart 8) where is also shown a chart by which the developed area ratio can be obtained for standard propellers of any pitch ratio and any given projected area ratio, and *vice versa*. On this same chart sheet is given a table of multipliers for obtaining the length of half-chord widths for different projected-area-ratio blades for any desired diameter of propeller, by means of which the projected-area forms can be laid down without the use of the chart of forms

FORM OF BLADE SECTIONS FOR STANDARD BLADES.

In the propellers designed according to the writer's method, the form of section existing in the propellers from whose data the charts were developed, has been adhered to. In these blades, the working face of the blade forms the nominal pitch surface, the blades in all cases being made with constant pitch. The thickness of the blade is built up on the back of this nominal pitch face. The form of the back is an arc of a circle and the edges are made as thin and sharp as possible without sacrificing durability to an extravagant degree.

Should the thickness, instead of all being applied to the back of the pitch surface, be divided on each side of it, the thrusts exerted by the blade per revolution will decrease and the tip speeds will increase, until the metal is equally divided on each side of the pitch face when the nominal pitch of the blade becomes equal to the true pitch. That this distribution of metal results in this equalization of true and nominal pitches is demonstrated by the fact that if a model blade of zero nominal pitch be

revolved in water and the thrusts measured, that blade having the metal divided equally on each side of the pitch plane will develop zero thrust, while all other blades will register thrust.

Should the leading side of the pitch face of a blade be thrown back in an attempt to fine up the entering edge the true pitch will be increased to a still greater extent over the nominal pitch, the speed of the blade will be lowered and the thrusts will increase. Should the after side of the blade be drawn back from the pitch plane there will, in the opinion of the writer, be a decided tendency toward the production of cavities in the wake of the blade, with resultant vibration.

EFFECT OF CHANGE OF BLADE FORM.

Should the forms of projected areas here advocated not be adhered to, the following results may be confidently looked for:

1. Broadening the blades at the tips.—Revolutions will be decreased, apparent slip will be decreased and thrusts will be increased.

2. Narrowing the blades at the tips.—Revolutions will be increased, apparent slip increased and thrusts decreased.

In the matter of relative weights for equal blade strengths the narrow-tipped blade has the advantage.

The writer wishes it to be distinctly understood that no claim is made that the forms advocated by him are necessarily those giving the maximum efficiency. He is of the opinion that equal efficiencies can be obtained with all shapes, if for each shape the proper diameter, pitch and surface have been provided for the absorption of the delivered power under the conditions in which the screw is operating. Each series of forms must, however, have its own particular factors of design if results in conformity with the computed performance are to be expected.

THICKNESS OF THE BLADE.

The fiber stress to be used in determining the thickness of the blade at the root depends upon the material of which the blade is to be made and the degree of approximation of the point of design to full overload conditions. The material usually used for propellers in the Naval Service is manganese bronze, and the specified ultimate tensile strength of the material is 60,000 pounds. Where the possible overload does not exceed ten per cent. a working stress of 10,000 pounds per square inch can be used with safety. This is the condition existing for Chart 5 of the Charts of Design. For propellers designed by Chart 6, where the power used in the calculations may be very much lower than the maximum power possible, this working stress should be reduced to about 6,000 pounds.

The formula used for the determination of blade thickness has been derived from Naval Constructor D. W. Taylor's work on "Resistance of Ships and Screw Propulsion," and is an adaptation of the formula proposed by him. The nomenclature and formulas are as follows:

T = Thickness of blade at and tangent to hub, additional thickness due to fillets being neglected. T should not exceed .2 W.

W = Width of blade tangent to hub.

A = (33,000 × I.H.P.) ÷ 2π × Revolutions × Number of blades = 5,252 I.H.P. ÷ R × N = Maximum indicated torque per blade, in foot pounds.

B = .31 × Diameter of screw, in feet = Mean arm.

C = A ÷ B = Resultant athwartship force on one blade, in foot pounds.

D = 12 × B − Radius of hub, in inches = Arm of athwartship force measured to root of blade.

E = C × D = Athwartship moment at root of blade, in inch pounds.

F = (33,000 × I.H.P.) ÷ Pitch, in feet × Revolutions × Number of blades = Indicated thrust per blade, in pounds.

G = .345 × Diameter of propeller, in inches = Mean arm of thrust, in inches.

H = G − Radius of hub, in inches = Arm of thrust measured to root of blade, in inches.

J = F × H = Fore-and-aft moment at root of blade, in inch pounds.

K = Circumference of hub, in feet ÷ Pitch, in feet = Tangent of angle between face of blade and center line of hub or fore-and-aft line tangent to surface of hub.

L = Sine of arc whose tangent is K.

M = Cosine of arc whose tangent is K.

N = L × J = Component of fore-and-aft moment normal to face of blade at root.

O = M × E = Same for athwartship moment.

P = N + O = Total moment at root of blade in inch pounds.

f = Fiber stress = $\begin{pmatrix} 10{,}000 \text{ for chart } 5 \\ 6{,}000 \text{ for chart } 6 \end{pmatrix}$ for manganese bronze of 60,000 T.S.

$$T = \sqrt{\frac{P \times 13.125}{W \times f}}.$$

CHARTS OF DESIGN.

These charts are five in number and are marked as numbers 3, 4, 5, 6 and 7.

Chart 3 is the Chart of Propulsive coefficients; 4 that of Tip Speeds. Although the lines of design from which Charts 5 and 6 are obtained are shown on Charts 3 and 4, these two latter charts are not used for actual design work but are used after the propeller has been designed, when it becomes desirable to lay down curves of expected performance through a range of speeds.

The actual Charts of Design are but two in number, namely 5 and 6. Before explaining these charts, it is advisable to explain the meanings of the different terms used in the charts.

DEFINITIONS OF TERMS AND ABBREVIATIONS.

I.H.P. = Indicated Horsepower of Main Engine on one screw.

S.H.P. = B.H.P. = .92 I.H.P. = Shaft or Brake Horsepower applied to the line shafting, and measured by torsion of shaft abaft the thrust bearing.

E.H.P. = Effective or Tow-rope Horsepower required to tow the hull at any given speed, the hull being fitted with all its appendages

$$= \frac{\text{Total E.H.P.}}{\text{Number of propellers}}.$$

P.C. = Actual Propulsive Coefficient $= \text{P.C.}_1 \div \text{C} = \text{P.C.}_1 \times \frac{1}{\text{C}} \left(\frac{\text{A}^s - s_1}{\text{A}^s(1-s_1)}\right)^s$.

P.C._1 = Propulsive Coefficient of the Propeller (Chart)

$$= \text{C} \times \frac{\text{E.H.P.}}{\text{I.H.P.}} = \text{C} \times \frac{\text{E.H.P.} \times .92}{\text{S.H.P.}}.$$

C = Thrust Deduction Factor for Standard Block Coefficient, from Chart 7.

D.A. = Disc area of propeller, and includes the area of the hub.

P.A. = Projected area of the propeller blades.

P.A. $\div$ D.A. = Projected area ratio.

H.A. = Helicoidal (developed) area of the propeller blades.

H.A. $\div$ D.A. = Developed area ratio.

P = Pitch of the propeller, in feet = Feet advance per revolution of the propeller if working in a non-yielding medium.

D = Diameter of the propeller, in feet.

R = Revolutions per minute of the propeller.

S = Speed of ship, in knots per hour.

s = Apparent slip of screw, expressed as a decimal

$$= [(\text{P} \times \text{R}) - (\text{S} \times 101.33)] \div \text{P} \times \text{R}.$$

Wake veloc. = Rate of motion of water directly at stern of vessel in same direction as the vessel is moving.

T.S. $= \pi \times \text{R} \times \text{D}$ = Tip speed of the screw, in feet per minute.

Total Indicated Thrust on one screw

$$= \frac{\text{I.H.P.} \times 33{,}000}{\text{P} \times \text{R}} = \text{I.T.} = \frac{\text{S.H.P.} \times 33{,}000}{.92 \times \text{P} \times \text{R}}.$$

Total Propulsive Thrust on one screw

$$= \text{P.T.} = \text{I.T.} \times \text{P.C.}_1.$$

Total Speed Thrust

$$= \frac{\text{I.H.P.} \times 33{,}000}{\text{S} \times 101.33} = \text{S.T.} = \frac{\text{S.H.P.} \times 33{,}000}{.92 \times \text{S} \times 101.33}.$$

Total Effective Thrust on one screw

$$= \text{E.T.} = \text{S.T.} \times \text{P.C.}_1.$$

$$\frac{\text{I.T.}}{\text{S.T.}} = \frac{\text{I.H.P.} \times 33{,}000}{\text{P} \times \text{R}} \times \frac{\text{S} \times 101.33}{\text{I.H.P.} \times 33{,}000} = \frac{\text{S} \times 101.33}{\text{P} \times \text{R}} = 1 - s.$$

$$\frac{\text{P.T.}}{\text{E.T.}} = \frac{\text{I.T.} \times \text{P.C.}_1}{\text{S.T.} \times \text{P.C.}_1} = \frac{\text{I.T.}}{\text{S.T.}} = 1 - s.$$

$$\text{I.T.}_d = \frac{\text{I.T.}}{\frac{\pi}{4} \times \text{D}^2 \times 144} = \text{Indicated thrust per square inch of disc area.}$$

$$\text{I.T.}_p = \frac{\text{I.T.}}{\text{P.A.} \times 144} = \frac{\text{I.T.}}{\frac{\text{P.A.}}{\text{D.A.}} \times \frac{\pi}{4} \text{D}^2 \times 144} = \text{Indicated thrust per square inch of projected area.}$$

$$\text{P.T.}_p = \text{I.T.}_p \times \text{P.C.}_1 = \text{Propulsive thrust per square inch of projected area.}$$

$$\text{E.T.}_p = \text{P.T.}_p \div (1 - s) = \text{Effective thrust per square inch of projected area.}$$

The formulas to use in calculation of the propeller coefficients take the following forms:

(1) $$\text{D} = \sqrt{\frac{291.8 \times \text{I.H.P.} \times \text{C}^2}{\text{P} \times \text{R} \times \text{I.T.}_d}} = \sqrt{\frac{291.8 \times \text{I.H.P.} \times \text{C}^2 \times \text{A}^4}{\text{P} \times \text{R} \times \text{I.T.}_d}}$$ } Full diameter. } Limited diameter.

(2) $$\text{D} = \sqrt{\frac{2.88 \times \text{E.H.P.} \times \text{C}^3}{\frac{\text{P.A.}}{\text{D.A.}} \times \text{E.T.}_p \times \text{S}}} = \sqrt{\frac{2.88 \times \text{E.H.P.} \times \text{C}^3 \times \text{A}^4}{\frac{\text{P.A.}_1}{\text{D.A.}_1} \times \text{E.T.}_p \times S_1}}$$

(3) $$\text{P} \times \text{R} = \frac{101.33 \times \text{E.T.}_p \times \text{S}}{\text{P.T.}_p}$$

(4) $$\text{P} = \frac{101.33\ \text{S} \times \text{E.T.}_p \times \pi\text{D}}{\text{T.S.} \times \text{P.T.}_p} = \frac{\text{P} \times \text{R} \times \pi\text{D}}{\text{T.S.}}$$

$$\text{R} = \frac{\text{T.S.}}{\pi\text{D}}.$$

The charts, as constructed, are based on a mechanical efficiency, for reciprocating engines of .92, and the propulsive coefficients only fit this efficiency. The relation between the I.H.P., S.H.P. and E.H.P. being expressed by

$$\text{I.H.P.} = \frac{\text{S.H.P.}}{.92} = \frac{\text{C} \times \text{E.H.P.}}{\text{P.C.}},$$

should the mechanical efficiency differ from .92, the relation between these powers must be corrected accordingly; thus suppose a mechanical efficiency of only .85 is expected, then $\text{I.H.P.} = \frac{\text{S.H.P.}}{.85} = \frac{\text{C} \times \text{E.H.P.} \times .92}{\text{P.C.} \times .85}$, and the I.H.P. to use in equation (1) would be only $\frac{85}{92}$ × this final I.H.P. of the main engines.

NUMBER OF BLADES.

The propeller charts have been developed from the performances of three-bladed propellers and therefore a correction must be applied should it be desired to use any other number of blades than three.

Should a four-blade wheel be required, multiply the I.H.P. of equation (1) or the E.H.P. of equation (2) by .865 and use the product in place of the actual I.H.P. or E.H.P. in calculating for the values of D, P and R. Multiply the projected area corresponding to these values of D, P and R by 4/3 to obtain the projected area of the four-bladed wheel.

Should we desire a two-bladed wheel, divide the I.H.P. or E.H.P. by .75 and multiply the projected area by $\frac{2}{3}$.

PROPELLERS OF REDUCED DIAMETER.

In many cases of propeller design, after computing a propeller to absorb a certain amount of power at a given number of revolutions, it will be found that, due to restrictions as to draught and hull clearance, the deduced diameter cannot be carried. It then becomes necessary to calculate a secondary screw of less diameter to absorb the designed power without exceeding the designed revolutions.

An investigation of the performances of propellers where the diameters, after preliminary trials, have been slightly reduced, accompanied by a slight reduction in pitch, shows, by comparison of performances before and after reduction, that where equal powers are applied to the screws the

$$\frac{\text{Revolutions after reduction}}{\text{Revolutions before reduction}} = \sqrt{\frac{\text{Diameter before reduction}}{\text{Diameter after reduction}}};$$

$$\text{or } \frac{R_2}{R_1} = \left(\frac{D_1}{D_2}\right)^{\frac{1}{2}} \text{ and that}$$

$$\frac{\text{Apparent slip after reduction}}{\text{Apparent slip before reduction}} = \frac{s_2}{s_1} = \left(\frac{D_1}{D_2}\right)^{4}.$$

The tip speeds after and before reduction will have the following ratios:

$$\frac{\text{T.S.}_2}{\text{T.S.}_1} = A = \frac{\pi R_2 D_2}{\pi R_1 D_1} = \frac{D_2}{D_1} \times \left(\frac{D_1}{D_2}\right)^{\frac{1}{2}} = \left(\frac{D_2}{D_1}\right)^{\frac{1}{2}};$$

$$\therefore D_2 = D_1 \times A^2, \text{ and } A = A^2 \times \frac{R_2}{R_1}; \therefore R_2 = R_1 \div A.$$

Assuming that the total projected areas are the same in the two cases, which can be done safely, as the primary screws used were narrow tipped so that the reduction in area due to cutting the diameters was negligible, and also, as a slight correction has been allowed in the revolution and slip exponents to allow for the slight increase

in the area of the secondary screw necessary to bring it equal to that of the primary one, the equation for projected area ratios takes the following form:

$$P.A._2 \div D.A._2 \ : \ P.A._1 \div D.A._1 \ :: \ D_1^2 \ : \ D_2^2;$$

$$P.A._2 \div D.A._2 = P.A._1 \div D.A._1 \times \left(\frac{D_1}{D_2}\right)^2 = P.A._1 \div D.A._1 \times \frac{1}{A^4}.$$

The apparent slips and the pitches of the primary and secondary propellers will bear the following ratios to each other:

$$\frac{s_2}{s_1} = \left(\frac{D_1}{D_2}\right)^4 = \frac{1}{A^8}\ ; \ \therefore s_2 = \frac{s_1}{A^8} \text{ and } P_2 = P_1 \times A.$$

DESCRIPTION OF THE CHARTS.

Chart 3. Chart of Propulsive Coefficients.—The ordinates of this chart are P.C.'s expressed as percentages, and the abscissas are pounds indicated thrust per square inch of disc area. The curves of P.C. are given for different values of P.A.÷D.A. and cover the practical ranges of this ratio.

It will be noted that all of these curves come tangent to an enveloping curve and then fall rapidly. Coinciding very closely with this enveloping curve is a curve for design which will be found again on Chart 5. Crossing these curves of P.C. at their points of maximum efficiency, is shown another curve which will be found reproduced on Chart 6 as a curve of design.

Chart 4. Curves of Tip Speeds.—The ordinates of this chart are tip speeds in feet per minute, while the abscissas are, as on Chart 3, pounds indicated thrust per square inch of disc area. On this chart as on Chart 3, the curves of tip speeds are shown for different values of P.A.÷D.A., and cover all practical values of that ratio. The curves are all tangent to an enveloping curve, as in Chart 3, and are crossed by two design lines of tip speeds, corresponding to the cross curves of design shown on Chart 3, and which are again shown on Design Charts 5 and 6.

These two charts, 3 and 4, are only used when laying down estimated curves of performance of a vessel. The data for design of propellers and estimate of power required are obtained by the use of either Chart 5 or 6, depending on the conditions under which the propeller operates.

Chart 5. Chart of Design.—High tip speeds, high speed of ship, maximum speed and power of vessel much greater than ordinary cruising speed and power.

Chart 6. Chart of Design.—Low tip speeds, low to moderate speed of ship, maximum speed of vessel not much greater than ordinary continuous speed, maximum power considerably in excess of power for continuous speed, this latter power and speed being used in the design. We may, however, design a propeller by each chart and then interpolate between for the final screw.

Propellers designed by Chart 5, have their points of maximum efficiency at considerably lower thrusts than at the point of design, while those designed by Chart 6 have their points of maximum efficiency coincident with the points of design.

Chart 5 is applicable to vessels where the power applied to the propellers is high, Chart 6 being used for those of low powers. In no case, however, should the propeller from Chart 5 be used when the resultant P. A. ratio is much less than .25.

The ordinates of these charts are P.C. in percentage; pounds thrust, $I.T._d$, $I.T._p$, $P.T._p$, and $E.T._p$; and tip speeds in feet per minute. The abscissas are values of P.A. ÷ D.A.

Commencing at the top of each chart, the upper curve is the design curve of P.C.; the second curve is a curve of $I.T._p$, which, however, is not used in the calculations; the third is a series of curves of $E.T._p$ for standard block coefficients ranging from .2 to .9; the fourth is a curve of $P.T._p$; the fifth of tip speeds; and, finally, the sixth curve is that of $I.T._d$.

To obtain data from these curves, suppose the vessel under consideration should have a block coefficient, standard by Chart 1, of .6, and we choose to enter the chart, say Chart 5, with a P.A. ÷ D.A. of .3; the data to be used in the calculations will be on the ordinate having the abscissa value of .3, and is as follows:

P.C. = .677, $E.T._p$ (from .6 curve) = 9.78, $P.T._p$ = 8.45, T.S. = 6,650, and should we desire to use equation (1) for the diameter, $I.T._d$ = 3.74. The value of C, the thrust deduction factor, is obtained for Standard B.C. = .6, from Chart 7. C = 1.04.

The apparent slip will equal $1 - \frac{P.T._p}{E.T._p} = 1 - \frac{8.45}{9.78} = .136.$

Should we use Chart 6, the data would become P.C. = .6935, $E.T._p$ = 6.4, $P.T._p$ = 5.79, T.S. = 5,600, $I.T._d$ = 2.5, and the apparent slip would equal $1 - \frac{5.79}{6.4} = .0953.$

REMARKS ON CHARTS 5, 6 AND 7.

An examination of these charts will show at once that they tie together, in a consistent manner, all the elements necessary to be taken into account in the design of a propeller, namely, the form of the ship with its influence on wake, flow of water to the screw and resultant slips, the effective and propulsive thrusts, the indicated thrusts and the rotary speed of the screw through the water. As any one of these elements is changed a corresponding change takes place in all the others to meet it. All are regarded as of equal importance.

While the values of P.A. ÷ D.A. have been taken for abscissas it is not because it is considered of more importance than the other elements, for such is not the case, as an examination of the charts reveals the fact that up to a value of .3 all projected area ratios deliver nearly equal propulsive coefficients. Beyond .3, however, the propulsive coefficient falls rapidly with an increase of this ratio. This equality of propulsive

coefficient for a wide variation of P.A.÷D.A. is often claimed, and it is seen that up to this value of .3 it is justified.

Later it will be shown how the value of P.A.÷D.A. varies in passing from Chart 6 to Chart 5, for the same conditions, and how little variation of P.C. results.

EFFECT OF CHANGE IN CHARACTERISTICS OF PROPELLER.

Suppose the propeller to be designed with blades separate from the hub. Call the designed pitch P, the diameter D and the blade area A, what will occur if we should change any one or more of these characteristics?

1. Pitch and blade area constant. Increase D. Effect: Efficiency increases, slip decreases, revolutions decrease. Decreasing D will have the opposite effect.

2. D and A remain constant. Increase P. Effect: Revolutions will decrease, slip will increase, efficiency will be increased at low thrusts and decreased at high ones. If P should be decreased, the opposite effects will result.

3. P and D remain constant. Increase A. Effect: Revolutions decrease, slip decreases, little effect on propulsive efficiency.

Finally, as a guide to judge the effect of general changes in the characteristics, should we take a constant form of hull, a constant E.H.P., and a constant speed, and should develop a series of propellers of different projected area ratios for these conditions, we would find that as we passed from the lowest to the highest projected area ratios, the diameter would decrease, the pitch would decrease, the revolutions would increase and the slips would increase.

The mistake is often made of supposing that a propeller is efficient because its apparent slip is low. This mistake is only justified to the extent that small projected area ratios have lower slips and have higher propulsive coefficients than have the higher projected area ratios, but the slips for a block coefficient of .2 will be considerably greater than for a block coefficient of .9, the projected area ratios and propulsive coefficients being the same for both blocks.

Thus, for P.A.÷D.A. of .25 and of .6, for B.C.=.2 and .9, the apparent slips would be as follows, from Chart 5:

B.C.	P.A.÷D.A.=.25	P.A.÷D.A.=.6
.2	.2019	.2552
.9	.0589	.1256

PROBLEMS MET WITH IN PROPELLER DESIGN.

These may be divided into two general classes: 1st, Problems of full diameter, and 2d, Problems of reduced diameter. The first are usually solved by the use of Chart 5, and the second by either 5 or 6, or by interpolation between them. These classes of problems may be still further divided into problems of sufficient and problems of insufficient data.

Two forms of computation are all that are needed, however, as where insufficient data is given, the missing data is not filled in and the propeller obtained is a guess at the best, and the client, if the work is being done for outside parties, should be so informed and no responsibility for results should be accepted.

In problems of insufficient data, the usual data supplied are as follows:

S = Estimated speed.
I.H.P. = Designed indicated horsepower of engines.
N = Number of propellers.
R = Designed revolutions for I.H.P. and S.
B.C. = Block coefficient of vessel.

$\frac{B}{L.W.L.}$ is not usually given but can generally be obtained.

Appendages fitted can usually be ascertained.

In problems of sufficient data, the following are the available data:

S = Designed speed.
E.H.P. = Effective horsepower (bare hull) for S.
I.H.P. = Designed power of engines [may be given or I.H.P. to give E.H.P.+ Appendages may be obtained from work].
N = Number of propellers fitted.
R = Designed revolutions for designed I.H.P.
B.C. = Block coefficient.
$\frac{B}{L.W.L.}$ = Ratio of greatest immersed beam to length on load water line.
I.M.S. = Coefficient of immersed midship section.

FORMS FOR COMPUTATION.

Each form should have at the top the name of the vessel, the name of the computer, the date and the available data, with the form of computation following; the number of the design chart used being also stated.

Form 1. Problems of Reduced Diameter.—Vessels to be designed for a constant cruising speed S. Diameter to be limited not to exceed a certain amount and revolutions fixed at a certain number. Usually Chart 6, except when abnormally large P.A. ÷ D.A. results.

(1) D = Maximum allowable diameter.
(2) R = Designed revolutions.
(3) T.S. = πRD = Tip speed of secondary screw.

(4) $T.S._1$ = Tip speed of primary screw. Must equal or be greater than T.S.

(5) $A = T.S. \div T.S._1$.

(6) A^4.

(7) A_s.

(8) $P.A._1 \div D.A._1$ = Projected area ratio corresponding to $T.S._1$ by Chart.

(9) $P.A. \div D.A. = (P.A._1 \div D.A._1) \div A^4$ = Projected area ratio.

(10) $I.T._d$ = Indicated thrust per square inch of disc area corresponding to $P.A._1 \div D.A._1$ by Chart.

(11) $P.T._p$ = Propulsive thrust corresponding to $P.A._1 \div D.A._1$ by Chart.

(12) $E.T._p$ = Effective thrust corresponding to $P.A._1 \div D.A._1$ and corrected block coefficient by Chart.

(13) $$\frac{P.T._p}{E.T._p} = 1 - s_1.$$

(14) $$s_1 = 1 - \frac{P.T._p}{E.T._p} = \text{Apparent slip of primary screw.}$$

(15) $$s = \frac{s_1}{A^s} = \text{Apparent slip of secondary screw.}$$

(16) $1 - s = 1 -$ Apparent slip of secondary screw, in decimal.

(17) S = Designed speed of vessel, in knots per hour.

(18) $S \times 101.33$ = Speed of vessel, in feet per minute.

(19) $$P \times R = P_1 \times R_1 = \frac{S \times 101.33}{1 - s} = \text{Pitch} \times \text{Revolutions of both primary and secondary screws.}$$

(20) $$S_1 = \frac{P_1 \times R_1 \times (1 - s_1)}{101.33} = \text{Speed to use in formulas.}$$

(21) E.H.P. = Effective horsepower for speed S.

(22) $$E.H.P._1 = E.H.P. \times \left(\frac{S_1}{S}\right)^3 = \text{Estimated effective horsepower for } S_1.$$

(23) C = Thrust deduction factor for standard block from Chart 7.

(24) $P.C._1$ = Propulsive coefficient for $P.A._1 \div D.A._1$ from Chart.

(25) $$I.H.P. = C \times \frac{E.H.P._1}{P.C._1} = \text{Estimated power required.}$$

(26) $$P.C. = \frac{E.H.P.}{I.H.P.} = \text{Actual estimated propulsive coefficient.}$$

(27) $$D=\sqrt{\frac{2.88\times E.H.P._1\times C^3\times A^4}{\frac{P.A._1}{D.A._1}\times E.T._p\times S_1}}=\sqrt{\frac{2.88\times E.H.P._1\times C^3}{\frac{P.A.}{D.A.}\times E.T._p\times S_1}}$$
$$=\sqrt{\frac{291.8\times I.H.P.\times C^2\times A^4}{P_1\times R_1\times I.T._d}}.$$

(28) $$P=\frac{P\times R\times \pi D}{T.S.}=\text{Pitch.}$$

(29) $$R=\frac{T.S.}{\pi D}=\text{Revolutions.}$$

Should a four-bladed propeller be required, use .865 $E.H.P._1$, or .865 I.H.P. in the equation for D, and for P.A. ÷ D.A. multiply equation (9) by 4/3.

Form 2. Problems of Full Diameter. Chart 5.

(1) P.A. ÷ D.A. = Projected area ratio from Chart.

(2) T.S. = Tip speed for P.A. ÷ D.A.

(3) $I.D._d$ = Indicated thrust per square inch of disc area corresponding to P.A. ÷ D.A.

(4) $P.T._p$ = Propulsive thrust per square inch of projected area corresponding to P.A. ÷ D.A.

(5) $E.T._p$ = Effective thrust for standard B.C. for P.A. ÷ D.A.

(6) $1-s=\frac{P.T._p}{E.T._d}=1-$ Apparent slip expressed as a decimal.

(7) S = Designed speed, in knots per hour.

(8) $S\times 101.33$ = Designed speed, in feet per minute.

(9) $P\times R=\frac{S\times 101.33}{1-s}=$ Pitch × Revolutions of the propeller.

(10) E.H.P. = Effective horsepower (all appendages) for speed S.

(11) C = Thrust deduction factor for standard block coefficient from Chart 7.

(12) $P.C._1$ = Propulsive coefficient from chart for P.A. ÷ D.A.

(13) $I.H.P.=C\times\frac{E.H.P.}{P.C._1}=$ Estimated I.H.P. for speed S.

(14) $S.H.P.=I.H.P.\times .92=$ Estimated shaft horsepower for speed S.

(15) $$D=\sqrt{\frac{2.88\times E.H.P.\times C^3}{\frac{P.A.}{D.A.}\times E.T._p\times S}}=\sqrt{\frac{291.8\times I.H.P.\times C^2}{P\times R\times I.T._d}}$$
= Diameter of propeller.

(16) $$P=\frac{P\times R\times \pi D}{T.S.}=\text{Pitch of propeller.}$$

(17) $$R=\frac{T.S.}{\pi D}=\text{Revolutions of propeller.}$$

(18) $$P.C.=\frac{E.H.P.}{I.H.P.}=\frac{P.C._1}{C}.$$

For a 4-bladed propeller correct E.H.P., I.H.P. and P.A.÷D.A. as before.

Should the problem be one of insufficient data, with either form, use the second formula for D, namely:

$$D=\sqrt{\frac{291.8\times I.H.P.\times C^2}{P\times R\times I.T._d}}\text{ for full diameter, or}$$

$$D=\sqrt{\frac{291.8\times I.H.P.\times C^2\times A^4}{P\times R\times I.T._d}}\text{ for reduced diameter.}$$

See table of Forms at end of article.

PROBLEMS IN DESIGN AND ESTIMATING.

Problem 1. Full Diameter. Maximum Thrust. Chart 5.

Battleship, speed 21 knots. E.H.P. (bare hull) = 16,600.

Length on load water line = 600 feet. Beam = 97 feet 6 inches.

$\frac{B}{L.W.L.}$ = .1625. Nominal block coefficient = .654.

Standard block coefficient = .625 (Chart 1).

Appendage resistance (two-shaft) = .176; (four-shaft) = .15 (Chart 2).

Coefficient M.S. = Standard.

Total resistance (two-shaft) = 16,600 × 1.176 = 19,528.

E.H.P. on one shaft = 9,764.

Total resistance (four-shaft) = 16,600 × 1.15 = 19,090.

E.H.P. on one shaft = 4,773.

C = 1.06; C^3 = 1.2.

Revolutions desired for two-shaft arrangement = 111; four-shaft = 220.

For two-shaft arrangement assume values of P.A.÷D.A. of .30, .31, .32, .33 and .34.

For four-shaft arrangement assume values of P.A.÷D.A. of .400, .425, .450, .475 and .500.

Use Form No. 3 of Formulas for Computation.

Two-Shaft.

P.A. ÷ D.A.	.30	.31	.32	.33	.34
T.S., feet	6650	6880	7080	7280	7470
$P.T._p$, pounds	8.45	8.57	8.68	8.78	8.87
$E.T._p$, pounds	9.72	9.90	10.03	10.15	10.30
$1 - s$	.8693	.8656	.8656	.8656	.8612
E.H.P., 1 shaft	9764	9764	9764	9764	9764
C × E.H.P.	10350	10350	10350	10350	10350
P.C.	.6775	.6730	.6680	.6630	.6575
2 × I.H.P.	30554	30758	30988	31222	31498
S	21	21	21	21	21
101.33 S	2127.93	2127.93	2127.93	2127.93	2127.93
P × R	2447.9	2458.4	2458.4	2458.4	2470.9
Diameter, feet	23.39	22.80	22.29	21.82	21.34
Pitch, feet	27.04	25.59	24.32	23.15	22.18
R.p.m.	90.51	96.07	101.1	106.2	111.4
2.88 log	0.45939	....	....	....	....
E.H.P.= log	3.98963	....	....	....	....
C^3 log	0.07593	....	....	....	....
(1) log	4.52495	4.52495	4.52495	4.52495	4.52495
P.A. ÷ D.A. log	9.47712	9.49136	9.50515	9.51851	9.53148
E.T. log	0.98767	0.99564	1.00130	1.00647	1.01284
S log	1.32222	1.32222	1.32222	1.32222	1.32222
(2) log	1.78701	1.80922	1.82867	1.84720	1.86654
D^2 = (1) — (2) log	2.73794	2.71573	2.69628	2.67775	2.65841
D log	1.36897	1.35787	1.34814	1.33888	1.32921
π log	0.49715	0.49715	0.49715	0.49715	0.49715
πD = (3) log	1.86612	1.85502	1.84529	1.83603	1.82636
P × R log	3.38879	3.39065	3.39065	3.39065	3.39285
P × R × πD log	5.25491	5.24567	5.23594	5.22668	5.21921
T.S. = (4) log	3.82282	3.83759	3.85003	3.86213	3.87332
P log	1.43209	1.40808	1.38591	1.36455	4.34589
R = (4) — (3) log	1.95670	1.98257	2.00474	2.02610	2.04696

Propeller for 111 Revolutions.

D=21′−5″.

P=22′−3″.

P.A.÷D.A.=.3392.

R.P.M.=111.

I.H.P.=31,475.

Four-Shaft.

P.A. ÷ D.A.	.400	.425	.45	.475	.500
T.S.	8550	9000	9500	10000	10550
$P.T._p$	9.30	9.43	9.55	9.67	9.78
$E.T._p$	10.88	11.05	11.23	11.41	11.61
$1 - s$	.8548	.8534	.8504	.8475	.8424
E.H.P.	4773	4773	4773	4773	4773
C × E.H.P.	5060	5060	5060	5060	5060
P.C.	.618	.602	.588	.573	.558
I.H.P.	8188	8405	8606	8831	9068
S.H.P. (total) = I.H.P. × 4 × .92 =	30132	30930	31670	32518	33370
S	21	21	21	21	21
101.33 × S	2127.93	2127.93	2127.93	2127.93	2127.93
P × R	2489.4	2493.5	2502.3	2510.8	2514.2
D	13.38	12.96	12.42	11.99	11.59
P	12.24	11.28	10.28	9.46	8.68
R	203.3	221	243.5	265.4	289.8
E.H.P. log	3.67879	....	....	....	....
2.88 log	0.45939	....	....	....	....
C^3 log	0.07593	....	....	....	....
(1) log	4.21411	4.21411	4.21411	4.21411	4.21411
P.A. ÷ D.A. log	9.60206	9.62839	9.65321	9.67669	9.69897
E.T. log	1.03663	1.03822	1.05038	1.05729	1.06483
S log	1.32222	1.32222	1.32222	1.32222	1.32222
(2) log	1.96091	1.98883	2.02581	2.05620	2.08602
D^3 = (1) — (2) log	2.25320	2.22528	2.18830	2.15791	2.12809
D log	1.12660	1.11264	1.09415	1.07896	1.06405
π log	0.49715	0.49715	0.49715	0.49715	0.49715
πD = (3) log	1.62375	1.60979	1.59130	1.57611	1.56120
P × R log	3.39609	3.39681	3.39834	3.39981	3.40040
P × R × πD log	5.01984	5.00660	4.98964	4.97592	4.96160
T.S. = (4) log	3.93197	3.95424	3.97772	4.00000	4.02325
P log	1.08787	1.05326	1.01192	0.97592	0.93835
R = (4) — (3) log	2.30822	2.34445	2.38642	2.42389	2.46205

Propeller for 220 Revolutions.

D = 12′ 11¼″.

P = 11′ 4½″.

P.A. ÷ D.A. = .4215.

R.P.M. = 220.

S.H.P. = 30,800.

Having obtained the propellers and the powers required to obtain the designed speed, it becomes necessary to lay down the "Curves of Performance" which we anticipate, that is, curves of "Speed—Power" and "Speed—Revolutions," as from these curves are obtained the expected fuel and water consumptions of the vessels for different speeds.

The charts to be used in obtaining these curves are Nos. 3 and 4, while the formulas are as follows:

T.S. = Tip speed for any thrust $I.T._d$ for P.A. ÷ D.A. of propeller, Chart 4.

$I.T._d$ = Indicated thrust per square inch disc area for T.S., Chart 4.

P.C. = Propulsive coefficient corresponding to P.A. ÷ D.A. and $I.T._d$, Chart 3.

C = Thrust deduction factor from Chart 7, corresponding to standard B.C.

P = Pitch of screw.

D = Diameter of screw.

R = Revolutions of screw per minute $= \frac{T.S.}{\pi D}$.

I.H.P. = Indicated horsepower for $I.T._d$ on one screw.

S.H.P. = .92 I.H.P.

E.H.P. = Effective horsepower for $I.T._d$.

$E.H.P. = \frac{I.H.P. \times P.C.}{C}$; Total $E.H.P. = n \times E.H.P.$

(1) $R = \frac{T.S.}{\pi D}$.

(2) $I.H.P. = \frac{P \times R \times D^2 \times I.T._d}{291.8 \times C^2} = \frac{P \times T.S. \times D \times I.T._d}{291.8 \times C^2 \times \pi} = \frac{P \times T.S. \times D \times I.T._d}{916.7 \times C^2}$.

n = Number of shafts.

$n \times I.H.P.$ = Total I.H.P.; $n \times S.H.P. = .92 \times$ Total I.H.P.

S = Speed corresponding to $n \times E.H.P.$ as obtained from model tank curves of "E.H.P.—Speed."

The work necessary to obtain the curves of performance for the two foregoing conditions follows:

Condition 1—Reciprocating Engines—Two Shafts. P.A.÷D.A.=.3392.

I.T.d.	T.S., Chart 4.	Pitch, feet.	Diameter, feet.	C.	I.H.P., Equation 2.	2×I.H.P.	P.C., Chart 8.	$\frac{2\times E.H.P.}{2\times I.H.P.\times P.C.}=\frac{}{C}$	Speed.	R.P.M.	Final P.C. =P.C.+C.
4.6	7450	22.25	21.4	1.06	15735	31470	.6577	19528	21	111	.6205
4.0	7100	22.25	21.4	1.06	13130	26260	.6725	16660	20.38	105.6	.6344
3.5	6760	22.25	21.4	1.06	10940	21880	.6820	14080	19.6	100.6	.6434
3.0	6350	22.25	21.4	1.06	8806	17612	.6860	11400	18.36	94.45	.6472
2.5	5900	22.25	21.4	1.06	6818	13636	.6870	8838	17.0	87.76	.6481
2.0	5300	22.25	21.4	1.06	4900	9800	.6850	6333	15.4	78.84	.6462
1.5	4500	22.25	21.4	1.06	3120	6240	.6825	4017	13.32	66.93	.6438
1.0	3350	22.25	21.4	1.06	1549	3098	.6810	1990	10.8	49.83	.6424

Condition 2—Turbine Engines—Four Shafts. P.A.÷D.A.=.4215.

I.T.d.	T.S., Chart 4.	Pitch, feet.	Diameter, feet.	C.	S.H.P.=Equation 2×.92.	4×S.H.P.	P. C., Chart 3.	$\frac{4\times E.H.P.=4\times}{S.H.P.\times P.C.}=\frac{}{.92\,C.}$	Speed.	R.P.M.	Final P. C. =P.C.+C.
6.5	8950	11.38	12.94	1.06	7700	30800	.6025	19090	21	220	.6340
6.0	8780	11.38	12.94	1.06	6929	27716	.6240	17732	20.7	216	.6398
5.5	8530	11.38	12.94	1.06	6170	24680	.6375	16136	20.33	209.8	.6537
5.0	8300	11.38	12.94	1.06	5458	21832	.6500	14556	19.89	204.2	.6665
4.5	8000	11.38	12.94	1.06	4735	18940	.6590	12800	19.18	196.8	.6758
4.0	7710	11.38	12.94	1.06	4056	16224	.6626	11024	18.3	189.7	.6795
3.5	7350	11.38	12.94	1.06	3384	13536	.6650	9228	17.32	180.8	.6819
3.0	6920	11.38	12.94	1.06	2731	10924	.6640	7436	16.28	170.2	.6809
2.5	6350	11.38	12.94	1.06	2088	8352	.6615	5664	14.95	156.2	.6783
2.0	5550	11.38	12.94	1.06	1460	5840	.6575	3937	13.32	136.5	.6742
1.5	4500	11.38	12.94	1.06	888	3552	.6525	2376	11.4	110.7	.6691
1.0	3250	11.38	12.94	1.06	428	1712	.6480	1136	9.2	79.95	.6645

The Speed–Revolutions, Speed–I.H.P., and Speed–S.H.P. curves obtained above are shown on Fig. 1, p. 32.

Problem 2. Reduced Diameter. Insufficient Data.

Motor Dory.—L.L.W.L.=18.4 feet. B=5 feet 1 inch.

B÷L.W.L.=.277. Nominal Block Coefficient=.467.

Draught=1 foot 2¼ inches. Single screw in wake of stern post.

Displacement=3,320 pounds. Standard B.C.=.76 (Chart 1, line Z).

Speed (estimated)=5.7 knots. S.H.P.=5.117.

Maximum allowed diameter of screw=16 inches. I.H.P.=$\frac{5.117}{.92}$=5.562.

Revolutions=500. Tip Speed=2,094.4 feet per minute.

Use Chart 6 and Form No. 2 of Formulas for Computation.

T.S.	2,094.4	2,094.4	2,094.4	2,094.4
$T.S._1$	2,094.4	2,200.0	2,400.0	2,600.0
A (log)	0.00000	9.97864	9.94082	9.90612
A^4 (log)	0.00000	9.91456	9.76328	9.62448
A^6 (log)	0.00000	9.82912	9.52656	9.24896
$P.A._1 \div D.A._1$	.142	.1475	.1575	.167
P.A. ÷ D.A.	.142	.180	.2717	.3965
$I.T._d$	.88	.93	1.03	1.13
$P.T._p$	4.33	4.44	4.58	4.73
$E.T._p$	4.58	4.68	4.84	5.00
$1 - s_1$	.9455	.9487	.9464	.9460
s_1	.0545	.0513	.0536	.0540
s	.0545	.07603	.15944	.30439
$1 - s$	.9455	.92397	.84056	.69561
S	5.7	5.7	5.7	5.7
101.33 S	577.58	577.58	577.58	577.58
P × R	610.87	625.11	687.13	830.32
I.H.P.	5.562	5.562	5.562	5.562
C	1.313	1.313	1.313	1.313
D (feet)	2.2854	1.9918	1.5167	1.1227
P (feet)	2.0941	1.8677	1.5632	1.3983
R.p.m.	291.7	334.7	439.55	593.85

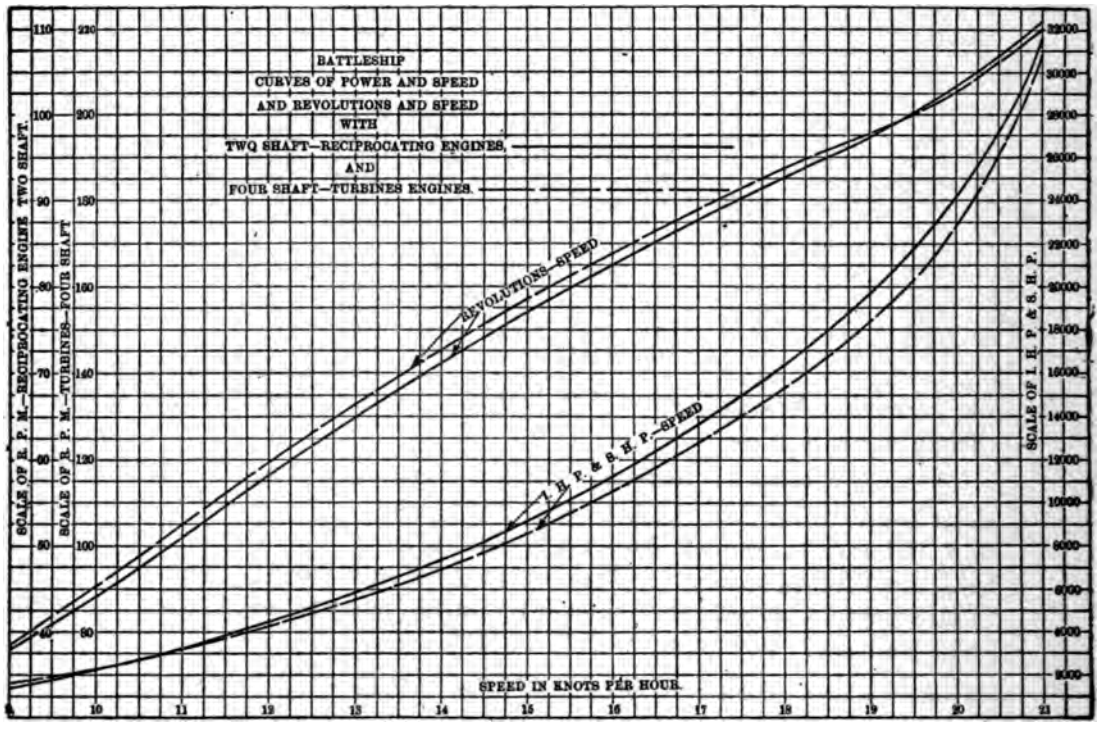

BATTLESHIP
CURVES OF POWER AND SPEED
AND REVOLUTIONS AND SPEED
WITH
TWO SHAFT—RECIPROCATING ENGINES,
AND
FOUR SHAFT—TURBINES ENGINES.
REVOLUTIONS—SPEED
I. H. P. & S. H. P.—SPEED
SCALE OF R. P. M.—RECIPROCATING ENGINES TWO SHAFT.
SCALE OF R. P. M.—TURBINES—FOUR SHAFT
SCALE OF I. H. P. & S. H. P.
SPEED IN KNOTS PER HOUR.

FIG. 1.

Plotting D, P and R on P.A.÷D.A. as abscissas, as shown on Fig. 2, we find the propeller for 500 revolutions to be as follows:

D=16 inches; P=17¾ inches; P.A.÷D.A.=.323; R=500; Speed=5.7 (estimated); Slip=21.7 per cent. Results obtained on trial: R=558; Speed=6.26.

The slips with propellers of reduced diameter are always higher than would be the case if the full diameter screw could be used. In some cases, where the decrease in

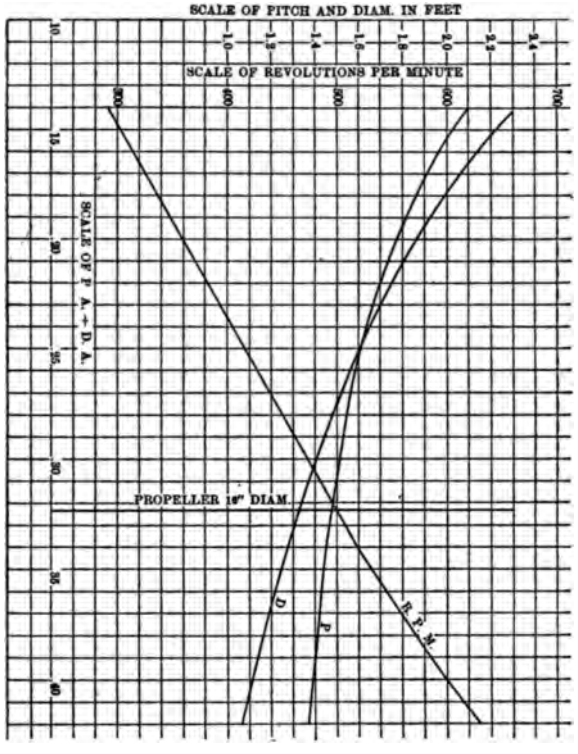


FIG. 2.

diameter is very great, the slip may be nearly 50 per cent., and the existence of such an amount of slip should not be taken as an accurate guide by which to condemn the propeller, for the propeller may be the best that can be fitted under the conditions existing, and the blame be laid on the conditions and not on the screw.

Problem 3. Problem of Reduced Diameter. Full Data.

Collier—Guaranteed continuous speed=14.3 knots. Use Chart 6. Revolutions for this speed, 90 per minute.

L.W.L.=528 feet; $\frac{B}{L.W.L.}$ =.1235; Nominal Block Coefficient=.726.

B = 65 feet 2¾ inches; Standard B.C. (Chart 1) = .615 (Line X).
Midship Section = .984; Twin Screw. E.H.P. (bare hull) = 3,430 for 14.3 knots.
Appendages { Struts and shafts. Rudder and post. Bilge keels.
E.H.P. (all appendages, Chart 2) = 3,430 × 1.129 = 3,872.
E.H.P. on one shaft = 1,936.
Maximum allowable diameter of screw = 16 feet 6 inches.
Tip speed at 90 revolutions per minute = 4,665.22.

Propeller, 3-bladed.

Use Form No. 1 of Formulas for Computation.

Problem 3.

T.S.	4,665.22	4,665.22	4,665.22	4,665.22
$T.S._1$	4,665.22	4,750	4,850	4,950
A (log)	0.00000	9.99218	9.98313	9.97426
A^4 (log)	0.00000	9.96872	9.93252	9.89704
A^8 (log)	0.00000	9.93744	9.86504	9.79408
$P.A._1 \div D.A._1$	.255	.259	.263	.268
P.A. ÷ D.A.	.255	.2784	.3072	.3397
$P.T._p$	5.62	5.65	5.68	5.70
$E.T._p$	6.15	6.18	6.21	6.23
$1 - s_1$	.9138	.9142	.9147	.9149
s_1	.0862	.0858	.0853	.0851
s	.0862	.0991	.1164	.1367
$1 - s$	.9138	.9009	.8836	.8633
S	14.3	14.3	14.3	14.3
101.33 S	1,449	1,449	1,449	1,449
P × R	1,585.74	1,608.4	1,640	1,678.5
S_1	14.30	14.511	14.803	15.155
E.H.P.	1,936	1,936	1,936	1,936
$E.H.P._1$	1,936	2,023.2	2,147.4	2,304.5
C	1.055	1.055	1.055	1.055
$E.H.P._{11} = C \times E.H.P._1$	2,042.5	2,134.5	2,265.5	2,431.2
$P.C._1$	.699	.6985	.698	.6977
I.H.P.	5,844	6,112	6,491.3	6,969.2
P.C.	.6626	.6335	.5965	.5556
D (feet)	17.086	16.556	16.036	15.588
P (feet)	18.246	17.932	17.709	17.62
R.p.m.	86.91	89.70	92.60	95.26

Plotting these results of D, P and R on P.A. ÷ D.A. as abscissas, we find the propellers to have the following characteristics:

D = 16 feet 6 inches;
P = 17 feet 11 inches;
P.A. ÷ D.A. = .281;
I.H.P. = 6,150.

It is usually considered that greater efficiency of propulsion can be obtained by the use of a single propeller. Whether this is true or not depends entirely upon the revolutions chosen and the diameter that can be carried, as is shown by the following problem where the conditions for the single screw are exaggerated in order to show the large differences that can be produced by bad choice of revolutions. The vessel is the same as that of the preceding problem.

Problem 4. Reduced Diameter. Sufficient Data.

Same vessel as in the preceding problem except single screw.

Revolutions = 50 per minute. Maximum diameter of screw = 27 feet 4 inches. Maximum tip speed = 4,288 feet per minute. Standard B.C. (Line Z, Chart 1) = .735. E.H.P. (Bare hull) = 3,450. C = 1.26, 4-bladed screw. E.H.P. (All appendages) Chart 2 = 3,450 × 1.029 = 3,550. (Use Chart 5.)

T.S.	4,288.	4,288.	4,288.	4,288.
$T.S._1$	4,288.	4,400.	4,500.	4,600.
A (log)	0.00000	9.98880	9.97904	9.96949
A^4 (log)	0.00000	9.95520	9.91616	9.87796
A^9 (log)	0.00000	9.91040	9.83232	9.75592
$P.A._1 \div D.A._1$	.204	.208	.212	.215
P.A. ÷ D.A.	.204	.2306	.2571	.2847
P.T.p	6.67	6.80	6.87	6.94
E.T.p	7.35	7.5	7.6	7.7
$1 - s_1$	.9075	.9067	.9040	.9013
s_1	.0925	.0933	.0960	.0987
s	.0925	.1147	.1383	.1735
$1 - s$	.9075	.8853	.8617	.8265
S	14.3	14.3	14.3	14.3
101.33 × S	1,449.0	1,449.0	1,449.0	1,449.0
P × R	1,596.7	1,636.77	1,681.58	1,753.24
S_1	14.3	14.646	15.002	15.594
.865 × E.H.P.	3,071.	3,071.	3,071.	3,071.
.865 × $E.H.P._1$	3,071.0	3,299.0	3,545.75	3,982.64
C	1.26	1.26	1.26	1.26
$P.C._1$	.6995	.6902	.6988	.6985
I.H.P.	6,395.	6,873.	7,391.	8,305.
D (feet)	28.87	26.53	25.52	25.73
P (feet)	33.77	33.01	32.68	33.05
R.P.M.	47.28	49.58	51.46	53.04
$P.A._{.7} \div D.A._{.7}$	.272	.3075	.3428	.3796

Plotting these points, we find the propeller necessary has the following characteristics:

D = 27.33 feet.
P = 32.9 feet.
P.A. ÷ D.A. = .315.
I.H.P. = 6,975.
E.H.P. = 3,550.
S = 14.3 knots.
P.C. = 50.9 per cent.

Comparing the propulsive coefficients obtained by the use of twin and single propellers in the foregoing two examples, we see that we obtain with

	P.C. (All Append.).	P.C. (Bare hull).
Single Screw, per cent.....	50.9	49.46
Twin Screws, per cent.....	62.96	56.11

Should a higher number of revolutions be chosen for the single screw and the reduction in diameter be less, these differences between the P.C.'s would be decreased. Should we be able to use the full diameter (primary), in both cases, there is no reason why equal P.C.'s on the hull with all appendages could not be obtained, and in such case, the single screw would be decidedly the better commercial proposition. This matter of choice of number of propellers will be discussed later.

Problem 5. Limitation of Speed due to Limitation of Propeller Diameter.

There were built for the U. S. Navy in 1912 two fuel-oil barges of the following dimensions: L.L.W.L.=160 feet; Greatest Immersed Beam=25 feet; Draught, mean=8 feet 3 inches; Displacement=835.5 tons.

The designed speed was six knots, and to obtain this speed a steam engine of 175 I.H.P. at 200 revolutions, driving a single screw, was fitted. As originally installed, a diameter of 6 feet 3 inches was the maximum that could be carried, and with a diameter of 6 feet 2 inches and a 4-bladed screw, the maximum speed obtained was 5.258 knots, the I.H.P. being 160, and the revolutions 207.6. The pitch of this propeller was 3 feet 6 inches.

Let us see what Chart 6 shows as to the speed possibility under these same conditions.

Data.

L.L.W.L.=160 feet; $\frac{B}{L.W.L.}$ =.15625; Standard B.C.=.895; C=1.59; B=25 feet.

Draught=8 feet 3 inches; Nominal B.C.=.886.

Displacement=835.5 tons.

There are available E.H.P. curves from which to obtain data.

T.S.	4,022.	4,022.	4,022.	4,022.
$T.S._1$	4,400.	4,800.	5,200.	5,600.
A (log)	9.96099	9.92320	9.88844	9.85625
A^4 (log)	9.84396	9.69280	9.55376	9.42500
A^5 (log)	9.68792	9.38560	9.10752	8.85000
$P.A._1 \div D.A._1$	.243	.261	.28	.3
P.A. ÷ D.A	.3481	.5418	.7822	1.1272
I.T.d	1.93	2.13	2.31	2.52
P.T.p	5.55	5.66	5.73	5.78
E.T.p	5.73	5.85	5.93	5.99
$1 - s_1$	.9686	.9675	.9663	.9655
s_1	.0314	.0325	.0337	.0345
s	.06442	.1337	.2631	.4873
$1 - s$	.93558	.8663	.7369	.5127
I.H.P.	160.	160.	160.	160.
$P.C._1$	.70	.6985	.697	.694
$E.H.P._{11} = I.H.P \times P.C._1$	112.	111.76	111.52	111.04
C	1.59	1.59	1.59	1.59
$E.H.P._1 = E.H.P._{11} \div C$	70.44	70.3	70.13	69.8
S_1 (Corresponding to $E.H.P._1$)	6.32	6.31	6.30	6.29
Log ($S_1 \times 101.33$)	2.80646	2.80577	2.80508	2.80439
Log (P × R)	2.82032	2.82012	2.81997	2.81964
S	6.105	5.65	4.805	3.34
D (feet)	7.474	5.98	4.893	4.041
P (feet)	3.86	3.087	2.525	2.084
R.P.M.	171.3	214.1	261.6	316.8
$P.A._{11} \div D.A._{11}$	.4641	.7224	1.043	1.503

Plotting these results, we find that the propeller should have the following dimensions, as given under the heading " Chart Propeller," but that six knots could not be made with this Chart Propeller.

	Chart Propeller.	Actual Propeller.
D=	6′ 2″	6′ 2″
P=	3′ 3″	3′ 6″
P.A.÷D.A.=	.678	.446
R=	207.6	207.6
S=	5.75	5.248
I.H.P.=	160.	160.
P.C.=	35.6 per cent.	26.55 per cent.

The pitch of the actual screw was too high and its area too small for the conditions existing and the diameter fitted. Having failed on trial, the line of shafting was lowered, permitting the fitting of a propeller of 6 feet 9 inches diameter. With this new condition, the following results were obtained: I.H.P.=177.64; Speed=6.235; Revolutions=206.27. The propeller was 3-bladed. The chart shows us as follows; Problem 6.

T.S.	4,375.	4,375.	4,375.	4,375.
$T.S._1$	4,500.	4,800.	5,000.	5,200.
A (log)	9.98777	9.95974	9.94201	9.92498
A^4 (log)	9.95108	9.83896	9.76804	9.69942
A^8 (log)	9.90216	9.67792	9.53608	9.39884
$P.A._1 \div D.A._1$	.248	.261	.271	.28
P.A. ÷ D.A.	.2776	.3782	.4623	.5588
$P.T._p$	5.58	5.66	5.70	5.73
$ET._p$	5.76	5.85	5.89	5.93
$1 - s_1$	.9687	.9675	.9670	.9663
s_1	.0313	.0325	.0330	.0337
s	.0392	.0682	.0960	.1342
$1 - s$	.9608	.9318	.9040	.8658
S	6.235	6.235	6.235	6.235
log (S × 101.33)	2.80058	2.80058	2.80058	2.80058
log (P × R)	2.81795	2.83126	2.84441	2.86316
S_1	6.286	6.474	6.67	6.95
$E.H.P._1$ (from E.H.P. curve)	69.5	73.1	76.8	82.78
C	1.59	1.59	1.59	1.59
$P.C._1$	.6995	.6985	.6975	.6965
I.H.P.	158.	166.4	175.	189.
D (feet)	8.947	7.687	6.997	6.447
P (feet)	4.225	3.743	3.511	3.378
R.P.M.	155.6	181.2	199.0	216.0

For Problem 6, the propeller will be as follows:

	Chart Propeller.	Actual Propeller.
D=	6′ 9″	6′ 9″
P=	3′ 5½″	3′ 6″
P.A.÷D.A.=	.494	.341
R.P.M.=	206.27	206.27
I.H.P.=	179.00	177.64
S=	6.235	6.235
Blades=	3.	3.

Comparing the above results with those of Problem 5, it is evident that the failure to make the contract speed under the latter condition was due almost entirely to being limited to too small a diameter of propeller and to a too high number of revolutions, as under Problem 6, where the revolutions were the same as under Problem 5, but with the diameter increased to 6 feet 9 inches, the speed was easily made. Attention is called to the close agreement of chart I.H.P. and actual I.H.P. for Problem 6. If there is any error existing in the charts in this high range of block coefficients and of C, it results in giving probably slightly greater projected areas than are required for the work to be accomplished, though the actual screws for both conditions produced heavy vibrations, thus indicating lack of surface.

Problem 7. Choice of Number of Propellers and Type of Machinery as Dictated by Economy of Propulsion.

Vessel having the following characteristics:

L.L.W.L.=416 feet; Greatest Immersed Beam=53 feet; Nominal B.C.=.63; $\frac{B}{L.W.L.}$ = .127; Draught = 25 feet; Standard B.C. = .55 (Twin Screw), = .675 (Single screw); Displacement=10,018 tons; C=1, (Twin screw)=1.06 (Single screw); Appendage Resistance = .1335, (Twin screws) = .0295 (Single screw); E.H.P. (Bare hull)=3,900; E.H.P. (All appendages)=3900×1.1335=4421 (Twin screw)=3900×1.0295=4015 (Single screw); Speed=16 knots.

Machinery proposed:

Case 1.	Case 2.	Case 3.
Reciprocating engines.	Reciprocating engines.	Turbine engines.
Single screw.	Twin screws.	Twin screws.
Revolutions, 68.	Revolutions, 95.	Revolutions, 240.

Find I.H.P. for cases 1 and 2 and S.H.P. for case 3 and the propulsive coefficients that may be expected in each case, also characteristics of the propellers for each case.

		Blades.	Maximum tip speed.
Case 1.	Maximum Diameter of propeller=22′–0″	4	4,712′
Case 2.	Maximum Diameter of propeller=15′–6″	3	4,626′
Case 3.	Maximum Diameter of propeller=11′–6″	3	8,671′

Case 1. Chart 5. Four-bladed Propeller.

T.S.	4,712	4,712	4,712	4,712
$T.S._1$	4,712	4,850	4,950	5,050
A (log)	0.00000	9.98747	9.97860	9.96992
A^4 (log)	0.00000	9.94988	9.91440	9.87968
A^5 (log)	0.00000	9.89976	9.82880	9.75936
$P.A._1 \div D.A._1$	.219	.225	.228	.2325
P.A. ÷ D.A.	.2919	.3366	.3701	.4089
P.T.p	7.03	7.16	7.22	7.32
E.T.p	7.85	7.98	8.10	8.22
$1-s_1$	.8956	.8972	.8914	.8905
s_1	.1044	.1028	.1086	.1095
s	.1044	.1295	.1611	.1906
$1-s$	.8956	.8705	.8389	.8094
S	16	16	16	16
S × 101.33 (log)	3.20986	3.20986	3.20986	3.20986
P × R (log)	3.25775	3.27009	3.28615	3.30170
S_1 (log)	1.20412	1.21724	1.23048	1.24559
E.H.P.	4,015	4,015	4,015	4,015
$E.H.P._1$ (log)	3.60369	3.64305	3.68277	3.72810
C	1.14	1.14	1.14	1.14
$P.C._1$	.6970	.6965	.6950	.6955
I.H.P.	6,567	7,195	7,890	8,764
P.C.	.611	.558	.5094	.458
D (feet)	23.21	22.1	21.56	21.09
P (feet)	28.01	27.44	27.78	28.16
R.p.m.	64.62	67.88	69.56	71.13

The desired propeller will have the following characteristics: D=22 feet; P=27.4 feet; R=68; P.A.÷D.A.=.34; P.C.=.554; I.H.P.=7,250.

Case 2. Chart 6. Twin Screws. Reciprocating Engines.

T.S.	4,626	4,622	4,626
$T.S._1$	4,626	4,700	4,800
A (log)	0.00000	9.99311	9.98397
A^4 (log)	0.00000	9.97244	9.93588
A^8 (log)	0.00000	9.94488	9.87176
$P.A._1 \div D.A._1$	.253	.256	.261
P.A. ÷ D.A.	.253	.2728	.3025
P.T.p	5.61	5.63	5.65
E.T.p	6.10	6.12	6.15
$1 - s_1$	.9197	.9191	.9186
s_1	.0803	.0809	.0814
s	.0803	.09185	.1044
$1 - s$	.9197	.90815	.8906
S	16	16	16
S × 101.33 (log)	3.20986	3.20986	3.20986
P × R (log)	3.24621	3.25170	3.26018
S_1 (log)	1.20412	1.20932	1.21757
E.H.P. (one screw)	2,211	2,211	2,211
$E.H.P._1$ (log)	3.34459	3.36019	3.38494
C	1	1	1
$P.C._1$	.699	.6985	.698
I.H.P.	6,326	6,562	6,952
P.C.	.699	.6737	.636
D (feet)	16.06	15.63	15.08
P (feet)	19.23	18.95	18.65
R.p.m.	91.7	94.23	97.61

The characteristics of the propeller will be: D=15 feet 6 inches; P=18 feet $10\frac{1}{2}$ inches; R=95; P.C.=.6658; I.H.P.=6,640; P.A.÷D.A.=.292.

Case 3. Turbine Engines. Twin Screws. Chart 5.

T.S.	8,671	8,400	8,200	8,000	7,700
P.A. ÷ D.A.	.411	.391	.381	.371	.352
P.T.p	9.36	9.24	9.17	9.10	8.96
E.T.p	11.11	10.97	10.89	10.81	10.62
$1 - s$	.8425	.8423	.8420	.8418	.8437
s	.1575	.1577	.1580	.1582	.1563
S	16	16	16	16	16
S × 101.33 (log)	3.20986	3.20986	3.20986	3.20986	3.20986
P × R (log)	3.28429	3.28439	3.28455	3.28465	3.28367
S_1 (log)	1.20412	1.20412	1.20412	1.20412	1.20412
E.H.P., 1 screw	2,211	2,211	2,211	2,211	2,221
$E.H.P._1$ (log)	3.34459	3.34459	3.34459	3.34459	3.34459
C	1	1	1	1	1
$P.C._1$	.612	.625	.631	.639	.65
I.H.P.	7,224	7,071	7,007	6,919	6,802
S.H.P.	6,646	6,505	6,447	6,366	6,258
D (feet)	9.336	9.633	9.794	9.962	10.32
P (feet)	6.508	6.934	7.225	7.534	8.094
R.p.m.	295.6	277.6	267.5	255.6	237.4

Propellers.

	Case 1.	Case 2.	Case 3.
D=	22′ 00″	15′ 06″	10′ 02⅝″
P=	27′ 05″	18′ 10½″	7′ 10″
P.A.÷D.A.=	.34	.292	.356
I.H.P.=	7,250	6,640	6,825
P.C.=	.554	.6658	.648
R=	68	95	240

Considering the efficiency of propulsion alone, the single-screw vessel would be the most extravagant to drive, which is contrary to the generally accepted idea. This inefficiency of the single screw is due entirely to the diameter being too restricted for the low number of revolutions. Should we desire to drive the vessel at the same cost in I.H.P. as for the twin-screw vessel the Propulsive Coefficient required would equal $4{,}015 \div 6{,}640 = .6047$; $P.C._1 = 1.14 \times .6047 = .6894$, and $E.H.P. = 4{,}015 \times 1.14 = 4{,}577$.

For full diameter this would correspond (Chart 6) to a $P.A. \div D.A. = .282$; $P.T._p = 8.18$; $E.T._p = 9.3$; Tip Speed = 6,220.

The screw corresponding to these conditions would be as follows:

D = 18′ 09″;
P = 17′ 05″;
R = 105.66;
P.A.÷D.A. = .376.

A decrease in revolutions will be followed by an increase in diameter and in pitch, a decrease in P.A.÷D.A. and an increase in propulsive efficiency until the limiting diameter of 22 feet has been reached. To further decrease the revolutions below those corresponding to a full diameter of 22 feet, results in a problem of reduced diameter and a rapid falling off in propulsive efficiency.

Estimated Curves of Performance for Propellers of Reduced Diameter.

In such cases there is not only a reduction of diameter and of tip speed below that given by the Charts, but the Propulsive Coefficient actually realized is considerably less than that given by the Charts, the reduction in Propulsive Coefficient depending directly upon the amount of diameter decrease and the thrust deduction factor C.

If we turn to Problem 7, cases 1 and 2, we will find a value P.C. and a value $P.C._1$, given for Propulsive Coefficients, where P.C. is the actual propulsive coefficient obtained and $P.C._1$ is the Corresponding Chart Propulsive Coefficient. The ratio between P.C. and $P.C._1$, that is

$$P.C. \div P.C._1 = e = \frac{(A^s - s_1)^s}{C \times [A^s(1-s_1)]^s} *$$

* This reduction factor for Propulsive Coefficient consists of two factors; $\frac{1}{C}$ = factor due to thrust deduction and $\left\{\frac{A^s - s_1}{A^s(1-s_1)}\right\}^s$ = factor due to limited diameter.

will be considered as applying to the designed propeller for all values of $I.T._d$ on Chart 3, so that to find the proper propulsive coefficient for a propeller of reduced diameter for any value of $I.T._d$, and for any value of $P.A.\div D.A.$, the following formulas are used:

$$I.H.P.=\frac{D^2\times P\times R\times I.T._d}{291.8\times C^2\times A^4}=\frac{P\times D\times T.S._1\times A\times I.T._d}{291.8\times\pi\times C^2\times A^4}=\frac{P\times D\times T.S._1\times I.T._d}{916.7\times C^2\times A^3};$$

$$R=\frac{T.S._1\times A}{\pi D}\text{ where }T.S._1=\text{Tip Speed for }I.T._d\text{ from Chart 4;}$$

$$P.C.=P.C._1\times e;$$

$$E.H.P.=I.H.P.\times P.C._1\times e;$$

Speed = Speed from E.H.P. Curve corresponding to E.H.P.

$P.A._1\div D.A._1$ (to enter Charts 3 and 4)

$= (P.A.\div D.A.)\times A^4$ for 3-bladed;

$= (P.A.\div D.A.)\times A^4\times\frac{3}{4}$ for 4-bladed wheels.

Differences in Characteristics of Propellers Designed from Charts 5 and 6.

To show the differences produced by designing propellers for the same conditions by Chart 5 and Chart 6, two problems will be solved and the resultant curves plotted on Fig. 3. This will illustrate how the diameter, projected area, and propulsive coefficient decrease and the slip increases for the same power and revolutions as we pass from Chart 6 condition to Chart 5 condition.

Suppose a vessel of Standard Block Coefficient of .5, Speed of 20 knots, E.H.P. (all appendages) applied to one screw = 1,000, the vessel being twin screw. Find series of propellers, revolutions varying, by Charts 6 and 5, for these conditions, also I.H.P. required. Full Chart Diameters can be carried.

The claim is often made that a large variation in projected area can be made without seriously affecting the efficiency of propulsion. That this claim is correct can readily be seen by consulting the Table of Comparison on Fig. 3, where are given columns of equal revolutions with corresponding pitches, diameters, projected-area ratios, and propulsive coefficients as obtained by Charts 5 and 6. It will be noted that Chart 6 gives for equal revolutions with Chart 5, slightly larger diameters, slightly lower pitches, much larger projected-area ratios, and practically equal propulsive coefficients. Suppose we take a compromise propeller between two propellers of equal diameter designed by the two Charts.

These propellers and the Compromise will be as follows:

	Chart 6.	Compromise.	Chart 5.
D=	8′ 00″	8′ 00″	8′ 00″
P=	9.01′	9.55′	9.9′
P.A.÷D.A.=	.340	.296	.252
P.C.=	.687	.689	.692
R=	250.	236.	223.

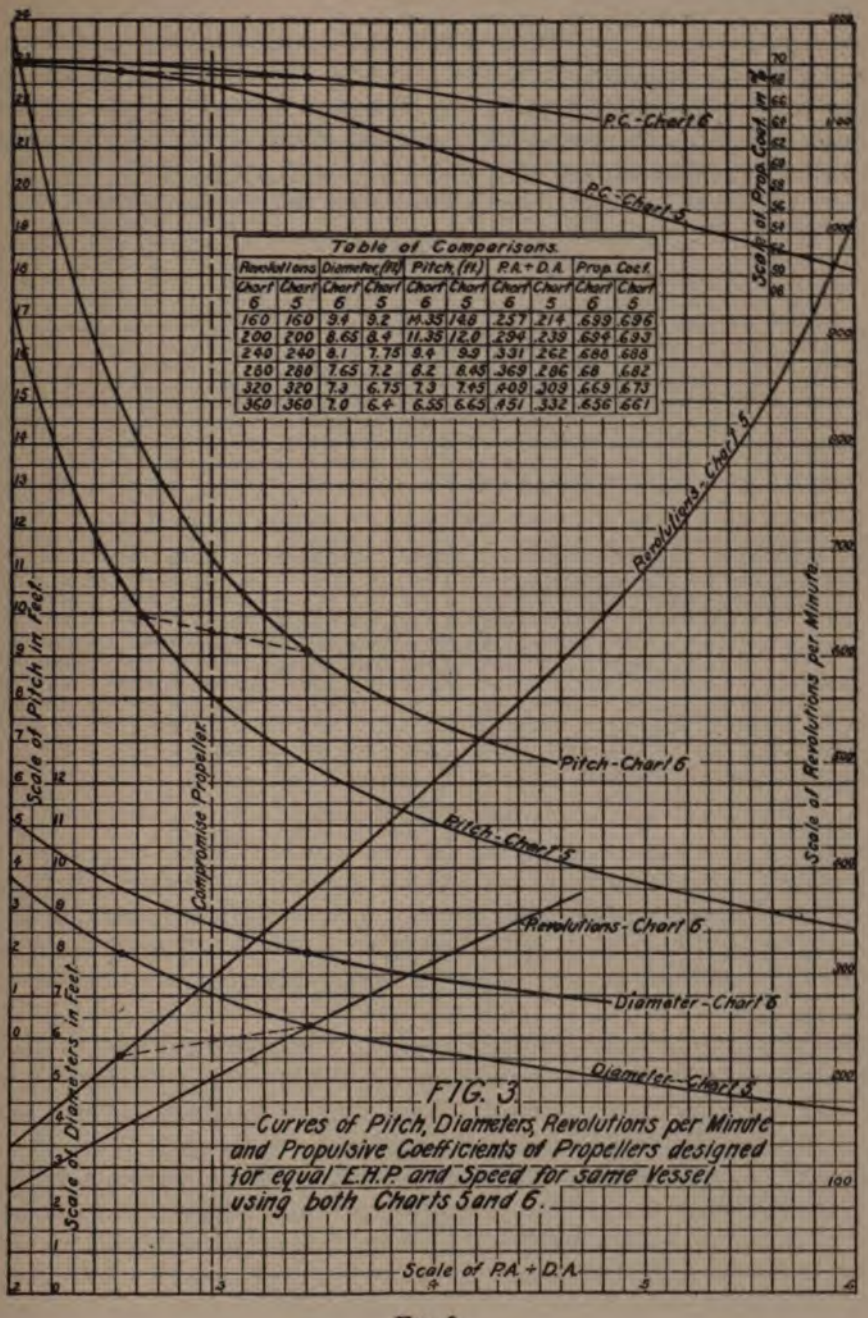


Table of Comparisons.

Revolutions		Diameter (ft)		Pitch (ft)		P.A. + D.A.		Prop. Coef.	
Chart 6	Chart 5	Chart 6	Chart 5	Chart 6	Chart 5	Chart 6	Chart 5	Chart 6	Chart 5
160	160	9.4	9.2	11.35	14.8	.257	.214	.699	.696
200	200	8.65	8.4	11.35	12.0	.294	.239	.694	.693
240	240	8.1	7.75	9.4	9.9	.331	.262	.686	.688
280	280	7.65	7.2	8.2	8.45	.369	.286	.68	.682
320	320	7.3	6.75	7.3	7.45	.409	.309	.669	.673
360	360	7.0	6.4	6.55	6.65	.451	.332	.656	.661

FIG. 3.

Curves of Pitch, Diameters, Revolutions per Minute and Propulsive Coefficients of Propellers designed for equal E.H.P. and Speed for same Vessel using both Charts 5 and 6.

FIG. 3.

Should we decrease the projected-area ratio of the compromise screw, retaining the original pitch, the propeller will speed up slightly, depending upon the amount of decrease and the slip will increase, while if the projected area be increased, the effect will be the opposite. In either case, the propulsive efficiency will probably not change more than the extreme limits given by the screws for Charts 5 and 6.

P.A. ÷ D.A.	T.S.	P.T.p.	E.T.p.	$1-s$.	E.H.P.	P.C.	2×I.H.P.	S.	101.33 S.	P × R (log).	D (feet).	P (feet).	R.P.M.
					BY CHART 6.								
.20	3,375	5.16	5.76	.8958	1,000	.7025	2,847	20	2,026.6	3.35453	11.18	23.54	96.09
.25	4,550	5.60	6.26	.8945	1,000	.6995	2,859	20	2,026.6	3.35516	9.593	15.01	151.
.30	5,580	5.79	6.50	.8908	1,000	.6935	2,884	20	2,026.6	3.35696	8.594	11.01	206.7
.35	6.550	5.81	6.60	.8802	1,000	.6845	2,922	20	2,026.6	3.36216	7.895	8.719	364.1
.40	7,130	5.75	6.56	.8765	1,000	.6720	2,976	20	2,026.6	3.36399	7.414	7.552	306.1
.45	7,800	5.61	6.41	.8752	1,000	.6560	3,049	20	2,026.6	3,36463	7.066	6.589	351.4
					BY CHART 5.								
.20	4,175	6.55	7.61	.8620	1,000	.698	2,865	20	2,026.6	3.37123	9.727	17.21	136.6
.25	5,480	7.67	8.97	.8551	1,000	.692	2,892	20	2,026.6	3.37472	8.013	10.89	217.7
.30	6,650	8.45	9.98	.8467	1,000	.677	2,954	20	2,026.6	3.37901	6.935	7.842	305.2
.35	7,650	8.95	10.69	.8372	1,000	.650	3,077	20	2,026.6	3.38391	6.204	6.167	392.5
.40	8,550	9.30	11.11	.8371	1,000	.619	3,231	20	2,026.6	3.38396	5.693	5.063	478.1
.45	9,500	9.55	11.47	.8326	1,000	.588	3,401	20	2,026.6	3.38630	5.276	4.246	573.2
.50	10,550	9.77	11.88	.8224	1,000	.558	3,584	20	2,026.6	3.39166	4.925	3.614	681.9
.55	11,780	9.95	12.38	.8037	1,000	.532	3,759	20	2,026.6	3.40165	4.599	3.092	815.4
.60	13,700	10.10	13.00	.7770	1,000	.506	3,953	20	2,026.6	3.41632	4.297	2.57	1,015.0
.625	15,300	10.17	13.36	.7612	1,000	.494	4,049	20	2,026.6	3.42524	4.153	2.27	1,173.0

Tug-Boat Propellers.

Professor Peabody, in writing on this subject, states as follows:

"A towboat, especially of the harbor type, has three various kinds of duty, namely: (1) To run free, passing from place to place, and perhaps seeking business in face of competition; (2) towing at a reduced speed, commonly at half speed or somewhat more, and (3) pulling or pushing a ship into her berth at a very low speed, or nearly without speed."

For such duties, the propeller adopted must be a compromise and must be designed to do its duty of minor importance without excessive sacrifice of its efficiency when performing the major important duty. Thus, a Navy sea-going tug boat must be able to steam economically with the fleet at the maximum cruising speed of the fleet, and at the same time must be fairly efficient when towing a vessel of moderate resistance.

Suppose the estimated maximum continuous speed of the tug boat is 14 knots, the I.H.P. of the engine, single screw, being, 1,800 and the revolutions for this power 120. The maximum diameter of the screw that can be fitted is 12 feet 9 inches. The propeller is required to work with maximum efficiency when its resistance has been in-

creased by a tow to such an extent that the maximum speed when developing full power at the designed revolutions is 13 knots. The screw to be 4-bladed.

The dimensions of the vessel are as follows: L.L.W.L.=175; B÷L.W.L.=.195; Greatest Immersed Beam=34 feet 1½ inches; Nominal B.C.=.531; Standard B.C. (Line Z)=.74; Draught (mean)=12 feet 6 inches; Displacement=1,135 tons; Coefficient of Immersed Midship Section=.875; C for Standard B.C.=1.27; Maximum diameter=12 feet 9 inches; R.p.m.=120; Maximum T.S.=4,807; I.H.P. =1,800.

Chart.	6	5	6	5	6	5
T.S.	4,807	4,807	4,807	4,807	4,807	4,807
$T.S._1$	4,807	4,807	4,950	4,950	5,050	5,050
A (log)	0.00000	0.00000	9.98726	9.98726	9.97858	9.97858
A^4 (log)	0.00000	0.00000	9.94904	9.94904	9.91432	9.91432
A^8 (log)	0.00000	0.00000	9.89808	9.89808	9.82864	9.82864
$P.A._1 \div D.A._1$	.261	.223	.269	.229	.273	.233
P.A. ÷ D.A.	.261	.223	.3025	.2575	.3325	.2838
P.T.	5.66	7.11	5.7	7.25	5.71	7.32
E.T.	6.02	7.88	6.08	8.00	6.10	8.11
$1 - s_1$	.9402	.9023	.9375	.9031	.9362	.9038
s_1	.0598	.0977	.0625	.0969	.0638	.0962
s	.0598	.0977	.07903	.1225	.09466	.1427
$1 - s$	.9402	.9023	.92097	.8775	.90534	.8573
S	13	13	13	13	13	13
S × 101.33 (log)	3.11968	3.11968	3.11968	3.11968	3.11968	3.11968
P × R (log)	3.14646	3.16433	3.15543	3.17643	3.16287	3.18655
S_1 (log)	1.11394	1.11394	1.12166	1.12643	1.12850	1.13688
I.H.P.	1,800	1,800	1,800	1,800	1,800	1,800
C	1.27	1.27	1.27	1.27	1.27	1.27
$I.T._d$	2.13	2.26	2.21	2.38	2.27	2.45
D (feet)	15.71	14.9	14.36	13.51	13.50	12.64
P (feet)	14.35	14.25	13.42	13.25	12.83	12.70
R.p.m.	97.65	102.7	106.6	113.3	113.4	121
$P.A._{.11} \div D.A._{.11}$	.381	.297	.4032	.3433	.4433	.3784
$P.C._1$ (charts)	.6985	.6965	.6976	.696	.6974	.695
e (log)	9.89620	9.89620	9.87280	9.85843	9.85258	9.82741
$P.C. = P.C._1 \times e$	.55	.5484	.5205	.5024	.4967	.4671

The propellers obtained will be as follows, the curves for Chart 6 propeller having been extended to a diameter of 12 feet 9 inches:

	Chart 6.	Chart 5.
Diameter	12′ 09″	12′ 09″
Pitch	12′ 06½″	12′ 08″
P.A.÷D.A.	.491	.374
R.p.m.	120	120
I.H.P.	2,286	2,286
Towing speed	13	13
P.C.	.480	.472

Should it be estimated that the engine has considerable reserve power the propeller from Chart 6 should be used; but if the engine is built with little reserve power the Chart 5 propeller should be adopted, as it would be considerably lighter than the other.

Methods of Estimating Power.

These are four in number, as follows:

1. The Admiralty Coefficient.
2. The Law of Comparison.
3. Independent Estimate.
4. Model Experiments.

The usual methods of estimating power, employed in the Bureau of Steam Engineering, are Nos. 1, 2 and 4.

Admiralty Coefficient.

The equation in which this coefficient occurs is

$$\text{I.H.P.} = \frac{D^{\frac{2}{3}} V^3}{K},$$

in which

I.H.P. = Indicated horsepower of the engine.
D = Displacement, in tons.
V = Speed, in knots per hour.
K = Admiralty Coefficient.

Where the Shaft Horsepower is given instead of the Indicated Horsepower, the equation may read:

$$\text{S.H.P.} = \frac{D^{\frac{2}{3}} V^3 \times .92}{K}.$$

This coefficient must be derived from some ship for which the displacement, power, and speed are known, and further, in order that a close agreement may be expected between the estimated speed and the actual trial speed of the ship, the coefficient must be derived from a ship that is geometrically similar to the ship under design, and which has the corresponding speed. These terms will be explained when the "Law of Comparison" is taken up. Furthermore, it is absolutely necessary that the conditions existing in the new ship are such as will permit the realization of an equal propulsive coefficient of propulsion with the compared vessel. Where these conditions exist, we may write

$$\text{E.H.P.} = \frac{D^{\frac{2}{3}} V^3 \times \text{P.C.}}{K}.$$

A moderate deviation in the first two requirements may not seriously affect the value of the method, but such is not the case with deviation from the third requirement.

Law of Comparison.

1. "Corresponding Speeds."—The corresponding speeds for similar ships are proportional to the square roots of their lengths.

2. "Displacements."—Similar ships have displacements proportional to the cubes of their lengths.

3. "Corresponding Speeds."—The corresponding speeds for similar ships are proportional to the sixth roots of their displacements.

4. "Horsepowers."—The horsepowers of similar ships at corresponding speeds are proportional to the seven-sixth powers of their displacements.

This rule (4), is not strictly correct, however, as the frictional resistance does not follow the law of mechanical similitude.

5. "Variation of Power with Speed."—Where the difference between the two speeds compared is small, we may assume that "The power for a ship is proportional to the cube of the speed," although this exponent may be widely departed from at high speeds.

6. "Variation of Power with Variation in Displacement."—For small changes in draught, we may assume that "The powers vary as the nth power of the Displacements," where n may vary from 2/3 for large ships of moderate speed to 1/6 for ships and boats of high speed.

In comparisons of hulls for similarity of form, Chart 1 should always be employed on account of the great influence of $\frac{B}{L.L.W.L.}$ and coefficient of immersed midship section on fullness of lines.

Independent Estimate.

The tow-rope resistance of a vessel is divided into two parts, surface or frictional resistance and residual resistance. The residual resistance is again divided into wave-making, eddy-making, and stream-line resistance.

The equation used for the calculation of frictional resistance is

$$R_f = fSV^n,$$

in which R_f is the force, in pounds, required to overcome the surface resistance, S is the wetted surface, in square feet, and V is the speed, in knots per hour. f and n are quantities taken from tables which can be obtained from any work on the "Resistance of Ships," and which are included here.

The equation used for finding the residual resistance is given as

$$R_w = \frac{bD^{\frac{2}{3}}V^4}{L},$$

where D, V and L are the displacement, in tons, the speed in knots per hour, and the length on the load water line, in feet. b is a numerical factor, having a value for long, fine ships of about .35; moderately fine ships, .40; ships broad in proportion to length but with fine ends, .45; freighters, .5. The values of b is also likely to be affected by speed, especially when the speed-length ratio is high.

Total Resistance.—As stated before, this is the sum of the two resistances, frictional and residual, and the equation for it is

$$R = R_f + R_w = fSV^n + \frac{bD^{\frac{2}{3}}V^4}{L}.$$

Using this equation in the estimation of the E.H.P., the equation for net E.H.P. takes the form

$$\text{E.H.P.} = 0.00307\left(fSV^{n+1} + \frac{bD^{\frac{2}{3}}V^5}{L}\right),$$

where the various letters have the same significance as before.

Wetted Surface.—This is determined from the lines of the ship and is a tedious operation. The surface is computed in square feet. For a preliminary design, the wetted surface may be computed by the equation $S = C\sqrt{DL}$, where D is the displacement, in tons, L the length on load-water line, and C a coefficient depending on the beam and draught.

Model Experiments.—The fourth method for determining power is by aid of model experiments in a towing basin. To illustrate the method, suppose that the tow-rope resistance for a paraffin model 20 feet long is 12.8 pounds, when towed at the speed corresponding to 25 knots for the full-sized vessel which has a length on load-water line of 700 feet, then

$$V_m : 25 :: 20 : 700 \therefore V_m = 4.23 \text{ knots.}$$

The wetted surface of the vessel is 67,540 square feet, therefore the wetted surface of the model:

$$S_m : 67{,}540 :: 20^2 : 700^2 \therefore S_m = 55.1 \text{ square feet.}$$

The friction factor and the exponent taken from Froude's tables are

$$f = 0.00834 \text{ and } n = 1.94;$$

therefore the frictional resistance is,

$$0.00834 \times 55.1 \times 4.23^{1.94} = 7.54 \text{ pounds.}$$

The total frictional resistance of the full-sized vessed, $f=0.00847$ and $n=1.825$, is $0.00847\times67{,}540\times25^{2.825}$ and the E.H.P. (frictional)

$$0.00307\times0.00847\times67{,}540\times25.^{2.825}=15{,}600.$$

Taking the frictional resistance of the model from the total tow-rope resistance of the model, gives for the residual resistance

$$12.8-7.54=5.26 \text{ pounds.}$$

The corresponding residual resistance for the ship is

$$R_w : 5.26 :: 700^3 : 20^3 \therefore R_w=225{,}500 \text{ pounds.}$$

At 25 knots the E.H.P. required to overcome this residual resistance will be $0.00307\times225{,}500\times25=17{,}310$.

The total E.H.P. will then be $15{,}600+17{,}310=32{,}910$.

In all the above methods, the results obtained are those for the bare hull only, and the appendage resistance increase called for by Chart 2 must be applied before we are in a position to correctly compute the propeller and the indicated and shaft horsepowers.

(Credit must be given Peabody's work on " Propellers " for the major part of the above sections on " Resistance of Ships."—C. W. D.)

Design of Propellers by Comparison.

Sometimes, in designing the propellers for a vessel, we may wish to obtain propellers which will give an equal propulsive efficiency with those fitted to an earlier vessel of similar form but of different size, and whose performance has been regarded as excellent.

To obtain such propellers, we can work directly with the Chart formulas, always remembering that, according to the Charts, for equal propulsive efficiencies the projected area ratios, thrusts (effective, propulsive and indicated), and tip speeds must remain equal. Bearing this in mind, and obtaining the ratios between the equations for diameter, pitch, and revolutions of the existing propellers and those of the propellers we are designing, the following equations result:

$$D_2=D_1\sqrt{\frac{E.H.P._2\times S_1}{E.H.P._1\times S_2}}=D_1\sqrt{\frac{I.H.P._2\times S_1}{I.H.P._1\times S_2}}=D_1\sqrt{\frac{S.H.P._2\times S_1}{S.H.P._1\times S_2}}.$$

$$P_2=P_1\sqrt{\frac{E.H.P._2\times S_2}{E.H.P._1\times S_1}}=P_1\sqrt{\frac{I.H.P._2\times S_2}{I.H.P._1\times S_1}}=P_1\sqrt{\frac{S.H.P._2\times S_2}{S.H.P._1\times S_1}}.$$

$$R_2=R_1\sqrt{\frac{E.H.P._1\times S_2}{E.H.P._2\times S_1}}=R_1\sqrt{\frac{I.H.P._1\times S_2}{I.H.P._2\times S_1}}=R_1\sqrt{\frac{S.H.P._1\times S_2}{S.H.P._2\times S_1}}.$$

Where

D_1 = Diameter of existing propeller;
D_2 = Diameter of propeller for new ship;
P_1 = Pitch of existing propeller;
P_2 = Pitch of propeller for new ship;
R_1 = Revolutions of existing propeller;
R_2 = Revolutions of propeller for new ship;
S_1 = Speed of existing vessel;
S_2 = Speed of new ship;
E.H.P.$_1$ = Effective horsepower for S_1 of existing vessel;
I.H.P.$_1$ = Indicated horsepower for S_1 of existing vessel;
S.H.P.$_1$ = Shaft horsepower for S_1 of existing vessel;
E.H.P.$_2$ = Effective horsepower for S_2 of new vessel;
I.H.P.$_2$ = Indicated horsepower for S_2 of new vessel;
S.H.P.$_2$ = Shaft horsepower for S_2 of new vessel.

Should the speeds used in the above be the corresponding speeds by Froude's Law of Comparison, where

$$S_2 = S_1 \left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{\frac{1}{6}} \text{ and}$$

$$\text{Power}_2 = \text{H.P.}_2 = \text{H.P.}_1 \left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{\frac{1}{2}},$$

the following forms will obtain:

$$D_2 = D_1 \left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{\frac{1}{3}} = D_1 \left(\frac{L_2}{L_1}\right)^{1} = D_1 \left(\frac{S_2}{S_1}\right)^{2} = D_1 \sqrt{\frac{\text{H.P.}_2 \times S_1}{\text{H.P.}_1 \times S_2}};$$

$$P_2 = P_1 \left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{\frac{1}{3}} = P_1 \left(\frac{L_2}{L_1}\right)^{2} = P_1 \left(\frac{S_2}{S_1}\right)^{4} = P_1 \sqrt{\frac{\text{H.P.}_2 \times S_2}{\text{H.P.}_1 \times S_1}};$$

$$R_2 = R_1 \left(\frac{\text{Displacement}_1}{\text{Displacement}_2}\right)^{\frac{1}{3}} = R_1 \left(\frac{L_1}{L_2}\right)^{1} = R_1 \left(\frac{S_1}{S_2}\right)^{2} = R_1 \sqrt{\frac{\text{H.P.}_1 \times S_2}{\text{H.P.}_2 \times S_1}}.$$

According to these formulas it appears that the "Law of Mechanical Similitude" does not apply to screw propellers, as the diameters are seen to vary approximately as the cubes of the speeds, while the pitches vary, with the same degree of approximation, as the fourth power.

CAVITATION.

Taylor, in his work on "Speed and Power of Ships," states as follows concerning this phenomenon:

"Nature of Cavitation.—The phenomenon known as cavitation has been given much attention, of late years, in connection with quick-running turbine-driven propellers. It appears to have been first identified upon the trials, in 1894, of the torpedo-

boat destroyer *Daring*, which had reciprocating engines. When driven at full power with the original screws this vessel showed very serious vibration, evidently due to some irregular screw action. The propulsive efficiency was poor, the maximum speed obtained being 24 knots for 3,700 I.H.P. and 384 revolutions per minute.

"Mr Sidney W. Barnaby, the engineer of the Thornycrofts, who built the *Daring*, came to the conclusion that at the high thrust per square inch at which the screws were working the water was unable to follow up the screw blades, and that 'the bad performance of the screws was due to the formation of cavities in the water forward of the screw, which cavities would probably be filled with air and water vapor.' So Mr. Barnaby gave the phenomenon the name of cavitation. The screws which gave the poor results had a diameter of 6 feet 2 inches, a pitch of 8 feet 7¾ inches and a blade area of 8.9 square feet. Various alternative screws were tried, and the trouble was cured by the use of screws 6 feet 2 inches in diameter, 8 feet 11 inches pitch and 12.9 square feet blade area. With these screws 24 knots was obtained with 3,050 I.H.P., and the maximum speed rose from 24 knots to over 29 knots."

For the *Daring* cavitation appeared to begin when the screw area was such that the thrust per square inch of projected area was a little over 11 pounds per square inch; 11¼ pounds is the figure given by Mr. Barnaby. "For a time it was thought that the thrust per square inch of projected area was a satisfactory criterion in connection with cavitation, and that the limiting thrust per square inch of projected area found on the *Daring* was generally applicable."

"This, however, is not the case. Greater thrusts have been successfully used and cavitation is liable to appear at much lower thrusts. In one case, within the author's experience, cavitation appeared when the thrust was about 5 pounds per equare inch of projected area, the tip speed being about 5,000 feet per minute, and in another when the thrust was about 7.5 pounds and the tip speed about 6,500 feet per minute. There is little doubt that the prime factors involved in cavitation are: (1) The speed of the blade through the water, which is evidently measured by the tip speed, and (2) the shape of the blade section."

While Mr. Taylor may be correct in his statements as to the prime factors involved in cavitation, the writer is of the opinion that as all of the elements of the propeller, namely, pitch, diameter, projected area and revolutions, and in addition, the form of the after body of the vessel behind which the propeller is working, each has its influence, it would be difficult, if not impossible, to differentiate between them as to their relative effect. Personally, he is of the opinion that the prime factor in regulating the thrusts that can safely be used on any particular propeller, is the form of the after body of the hull. As the after body fines, the thrust may be increased, and *vice versa*. Thus Chart 5 is derived from the actual performance of numerous vessels, and the values of $E.T._p$ given for the different standard block coefficients are con-

sidered as those which can safely be used without noticeable cavitation occurring. It is estimated that we may exceed these Chart 5 thrusts by at least 10 per cent. without the vibration from cavitation becoming excessive.

The thrusts given on Chart 5 are unaffected by either thrust deduction or reduction of diameter. Considering these two latter elements, there result three formulas for obtaining effective thrust, viz:

(1) Net Effective Thrust

$$= \text{E.T.}_p = \frac{2.88 \times \text{E.H.P.}}{(\text{P.A.} \div \text{D.A.}) \times \text{S} \times \text{D}^2}. \quad \text{Full diameter and no thrust deduction.}$$

(2) Net Effective Thrust

$$= \frac{\text{E.T.}_p}{C^3} = \frac{2.88 \times \text{E.H.P.}}{(\text{P.A.} \div \text{D.A.}) \times \text{S} \times \text{D}^2}. \quad \text{Full diameter and thrust deduction.}$$

(3) Net Effective Thrust

$$= \frac{\text{E.T.}_p}{C^3} \times \left\{\frac{A^3 - s_1}{A^3(1 - s_1)}\right\}^2 = \frac{2.88 \times \text{E.H.P.}}{(\text{P.A.} \div \text{D.A.}) \times \text{S} \times \text{D}^2}. \quad \text{Reduced diameter and thrust deduction.}$$

Thus, by (1), the E.T.$_p$'s obtained would be those which are shown on Chart 5, while by (2), they would be reduced by the cube of the thrust deduction factor, and by (3), in addition to the reduction due to thrust deduction, the diameter reduction factor also enters.

Let us take a value of P.A.$\div$D.A.$=.3$ and also$=.6$ on Chart 5 and tabulate the E.T.$_p$'s for the Block Coefficients from .9 to .2, both with and without the thrust deduction factor, and allow an overload of 10 per cent.; the following results:

Standard Block Coefficients.

P.A. $\div$ D.A.	..	.2	.3	.4	.5	.6	.7	.8	.9
.3	(1)*	10.62	10.41	10.19	9.98	9.77	9.57	9.33	9.02
	10 p. c.	11.68	11.45	11.20	10.98	10.75	10.53	10.23	9.92
	(2)*	10.62	10.41	10.19	9.98	8.73	5.70	3.43	2.20
	10 p. c.	11.68	11.45	11.20	10.98	9.60	6.27	3.77	2.42
	T.S.	6,650	6,650	6,650	6,650	6,650	6,650	6,650	6,650
.6	(1)*	13.56	13.39	13.21	13.00	12.77	12.45	12.01	11.55
	10 p. c.	14.92	14.72	14.53	14.30	14.05	13.70	13.21	12.71
	(2)*	13.56	13.39	13.21	13.00	11.40	7.41	4.42	2.81
	10 p. c.	14.92	14.72	14.53	14.30	12.54	8.15	4.86	3.09
	T.S.	13,600	13,600	13,600	13,600	13,600	13,600	13,600	13,600

* (1) Without thrust deduction. (2) With thrust deduction.

From this table, based on actual performances, it is seen that with a P.A.÷D.A. of .3 and a constant tip speed of 6,650 feet per minute, the effective thrusts for safe design, taking into account the thrust deduction, vary from 2.2 pounds per square inch of projected area, the vessel having standard B.C. of .9, to 10.62 pounds with a standard B.C. of .2, and that these values of thrust can be increased to 2.42 and 11.68 pounds respectively without the vibration becoming excessive.

Table of C, for Wetted Surface.

B+H	C	B+H	C	B+H	C
2.0	15.63	2.5	15.50	3.0	15.62
2.1	15.58	2.6	15.51	3.1	15.66
2.2	15.54	2.7	15.53	3.2	15.71
2.3	15.51	2.8	15.55	3.3	15.77
2.4	15.50	2.9	15.58	3.4	15.83

B = beam. H = draught.

FROUDE'S SURFACE-FRICTION CONSTANTS.

Given by Taylor.

Surface-Friction Constants for Paraffin Models in Fresh Water. Exponent n=1.94.

Length.	Coefficient.	Length.	Coefficient.	Length.	Coefficient.
Feet.		*Feet.*		*Feet.*	
2.0	0.01176	10.0	0.00937	14.0	0.00883
3.0	0.01123	10.5	0.00928	14.5	0.00877
4.0	0.01083	11.0	0.00920	15.0	0.00873
5.0	0.01050	11.5	0.00914	16.0	0.00864
6.0	0.01022	12.0	0.00908	17.0	0.00855
7.0	0.00997	12.5	0.00901	18.0	0.00847
8.0	0.00973	13.0	0.00895	19.0	0.00840
9.0	0.00953	13.5	0.00889	20.0	0.00834

Surface-Friction Constants for Painted Ships in Sea Water. Exponent n=1.825.

Length.	Coefficient.	Length.	Coefficient,	Length.	Coefficient.
Feet.		*Feet.*		*Feet.*	
8	0.01107	40	0.00981	180	0.00904
9	0.01177	45	0.00971	200	0.00902
10	0.01161	50	0.00963	250	0.00897
12	0.01131	60	0.00950	300	0.00892
14	0.01106	70	0.00940	350	0.00889
16	0.01086	80	0.00933	400	0.00886
18	0.01069	90	0.00928	450	0.00883
20	0.01055	100	0.00923	500	0.00880
25	0.01029	120	0.00916	550	0.00877
30	0.01010	140	0.00911	600	0.00874
35	0.00993	160	0.00907		

Surface-Friction Constants. Exponent, 1.825.

Given by Denny.

Length.	Coefficient.	Length.	Coefficient.	Length.	Coefficient.
Feet.		*Feet.*		*Feet.*	
40	0.00996	260	0.00870	550	0.00853
60	0.00957	280	0.00868	600	0.00850
80	0.00933	300	0.00866	650	0.00848
100	0.00917	320	0.00864	700	0.00847
120	0.00905	340	0.00863	750	0.00846
140	0.00896	360	0.00862	800	0.00844
160	0.00889	380	0.00861	850	0.00842
180	0.00884	400	0.00860	900	0.00841
200	0.00879	420	0.00859	950	0.00840
220	0.00876	450	0.00858	1,000	0.00839
240	0.00872	500	0.00855		

TIDEMAN'S SURFACE-FRICTION CONSTANTS.

Derived from Froude's Experiments.

Surface-Friction Constants for Ships in Salt Water of 1.026 Density.

Length of ship in feet.	Iron bottom clean and well painted.		Copper or zinc sheathed.			
			Sheathing smooth and in good condition.		Sheathing rough and in bad condition.	
	f	*n*	*f*	*n*	*f*	*n*
10	0.01124	1.8530	0.01000	1.9175	0.01400	1.8700
20	0.01075	1.8490	0.00990	1.9000	0.01350	1.8610
30	0.01018	1.8440	0.00903	1.8650	0.01310	1.8530
40	0.00998	1.8397	0.00978	1.8400	0.01275	1.8470
50	0.00991	1.8357	0.00976	1.8300	0.01250	1.8430
100	0.00970	1.8290	0.00966	1.8270	0.01200	1.8430
150	0.00957	1.8290	0.00953	1.8270	0.01183	1.8430
200	0.00944	1.8290	0.00943	1.8270	0.01170	1.8430
250	0.00933	1.8290	0.00936	1.8270	0.01160	1.8430
300	0.00923	1.8290	0.00930	1.8270	0.01152	1.8430
350	0.00916	1.8290	0.00927	1.8270	0.01145	1.8430
400	0.00910	1.8290	0.00926	1.8270	0.01140	1.8430
450	0.00906	1.8290	0.00926	1.8270	0.01137	1.8430
500	0.00904	1.8290	0.00926	1.8270	0.01136	1.8430

Where the P.A.÷D.A. has been increased to .6, these safe thrusts become 2.81 and 13.56 pounds, and the overloads 3.09 and 14.92 pounds respectively, the tip speeds for the safe loads having a constant value of 13,600 feet per minute.

In many cases where vibration is charged to cavitation, it may be due to an entirely different cause, that is, too close proximity of the blade tips to the hull or too close proximity of the leading edges of the blades to the after edge of the struts or stern post. Such vibration is generally localized, however, and becomes unnoticeable when the observer moves from the locality to which the vibration is transmitted by the hull structure.

Dyson's Formulas for Computation of Screw Propellers.

Symbols, definitions and formulas for propeller design.		Problems where diameter is limited.		Problems where diameter is unlimited.	
		Full data for design.	Insufficient data for design.	Full data for design.	Insufficient data for design.
		Hull data... {"L.L.W.L." / "B" / "Displ." | "Block Coef." "B" / L.W.L.} "Revs."—"Diam."—"E.H.P. (bare hull)." "S."—"Stand.B.C."—"C."—Form No. 1.	Hull data... {"L.L.W.L." / "B" / "Displ." | "Block Coef." "B." / L.W.L.} "Revs."—"Diam."—"S."—"Estimated I.H.P." "Stand.B.C."—"C."—Form No. 2.	Same hull data as Form No. 1. Diameter and revolutions unlimited. Form No. 3.	Same hull data as Form No. 2. Diameter and revolutions unlimited. Form No. 4.
$T.S.$ = Tip speed allowed	1	log (πR.D.) (allowed R. and D.) = log (1).	log (1)		
$T.S._1$ = or > T.S. = Tip speeds from Charts 5 and 6.	2	log $T.S._1$ = log (2)	log (2)	log (2)	log (2).
$A.$ = T.S. ÷ $T.S._1$	3	log (1) — log (2) = log (3)	log (3)		
$A.^4$	4	4 log (3) = log (4)	log (4)		
$A.^8$	5	8 log (3) = log (5)	log (5)		
$P.A._1 \div D.A._1$ = Projected area ratio from Charts 5 or 6 corresponding to $T.S._1$.	6	log (6)	log (6)	log (6)	log (6).
$P.A. \div D.A.$ = ($P.A._1 \div D.A._1$) ÷ $A.^4$ = Projected area ratio of final 3-bladed screw.	7	log (6) — log (4) = log (7)	log (7)		
$I.T._d$ = Indicated thrust, in pounds, from Charts 5 or 6 for $T.S._1$.	8		log (8)		log (8).
$P.T._p$ = Propulsive thrusts for $T.S._1$	9	log (9)	log (9)	log (9)	log (9).
$E.T._p$ = Effective thrusts for $T.S._1$	10	log (10)	log (10)	log (10)	log (10).
$1 - s_1 = P.T._p \div E.T._p$	11	log (9) — log (10) = log (11)	log (11)	log (11)	log (11).
$s_1 = 1 - \frac{P.T._p}{E.T._p}$ = Initial apparent slip	12	(12)	log (12)	(12)	(12).
$s = \frac{s_1}{A^8}$ = Actual apparent slip	13	log (12) — log (5) = log (13)	log (13)	(13) = (12)	(13) = (12).
$1 - s$	14	(14)	log (14)	log (14) = log (11).	log (14) = log (11).
S = Designed speed of ship	15	log (15)	log (15)	log (15)	log (15).
101.33 S = Speed, feet per minute	16	2.00574 + log (15) = log (16)	log (16)	log (16)	log (16).